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AFML-TR-69-249

DYNAPAK
1 - 1/2 MILLION FOOT POUND
FORGING MACHINE

M. Chanin
J. Koskoris
General Dynamics
Electro Dynamics Operation

TECHNICAL REPORT AFML-TR-69-249

September 1969

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FOREWORD

This report was prepared by General Dynamics Corporation, Electro Dynamics Division for the United States Air Force under Contract F33615-68-C-1578. The contract was initiated under Manufacturing Technology Division Project 173-8, "Design for 1 - ½ Million Foot Pound Dynapak Forging Machine". The work was administered under the direction of the Air Force Materials Laboratory, Manufacturing Technology Division, Wright-Patterson Air Force Base, Ohio with Mr. L. C. Polley as project engineer.

This project was under the direction of Mr. Milt Chanin, Chief Engineer, DYNAPAK, Electro Dynamic Division, General Dynamics Corporation, Avenel, New Jersey. He was assisted in the investigation design project by Mr. James Koskoris, Senior Project Engineer.

The project was accomplished as part of the Air Force Manufacturing Methods Program, the primary objective of which is to implement, on a timely basis, manufacturing processes, techniques, and equipment for use in economical production of USAF materials and components. This program encompasses the following technical areas:

Metallurgy	-	Rolling, Forging, Extruding, Casting, Drawing Powder Metallurgy, Composites
Chemical	-	Propellants, Coatings, Ceramics, Graphites, Non-metallics
Electronic	-	Solid State, Materials & Special Techniques, Thermionics
Fabrication	-	Forming, Material Removal, Joining, Components

Suggestions concerning additional Manufacturing Methods required on this or other subjects will be appreciated.

This Technical Report has been reviewed and is approved.

H. A. JOHNSON
Chief, Materials Processing Branch
Manufacturing Technology Division

ABSTRACT

A manufacturing methods requirement for this design study and engineering plans was established by Award/Contract No. F33615-68-C-1578 dated 22 April 1968. The 1-1/2 Million Foot Pound High Energy Rate Forging Press which was designed under this contract is a pneumatic-mechanical type press of true counter-blow design. The machine incorporates mechanical, hydraulic and electrical systems.

Two (2) integral opposing rams, weighing 73,000 pounds each, and having a maximum impact velocity of 630 in./sec. create the rated energy of 1,500,000 foot pounds. The two rams are proportioned for extreme stiffness in order to support large tooling requirements. Complete tooling interchangeability between the upper and lower rams is provided.

The utilization of an IBM 1800 computer for design studies and for stress analysis of key mechanical components was unique for the press industry and enabled us to include in the programs, the effect of more variables than can usually be accounted for.

The design incorporates unique rotating hydraulic jack assemblies for the external ram recocking system.

The hydraulic portion of the pneumatic high velocity machine restores the energy released by the machine. The hydraulic system is composed of two (2) 100 HP motor driving four (4) 35 GPM pumps which provide fluid for a high performance accumulator system. This system is capable of sustaining three (3) successive blows of the rams within several seconds. All functions are interlocked to provide maximum smoothness of hydraulic fluid delivery with minimum inertia shock.

The electrical system consists of conventional electrical circuitry monitored and controlled from a single console deck. All automatic events are under control of the operator. Each of the automatic events is sequenced and functioned by separate stepping relays which insures that all programmed events take place chronologically.

The machine design provides a total controlled energy output of 1,500,000 foot pounds with the capability of utilizing this energy in multiple blows.

The counter-blow principle and identical rams provides versatility in tooling capabilities.

The composite design of the machine emphasizes simplicity, flexibility, safety, minimum maintenance requirements and increased production capability.

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LIST OF ILLUSTRATIONS

1-1/2 MILLION FOOT POUND FORGING MACHINE

MODEL 2436

PROJECT NO. TB-8-MMP-254/123-8

I. INTRODUCTION

A. General

1. Purpose - The purpose of this program was to develop and establish complete design and engineering plans for a larger high velocity (HERF) type machine having controlled energy and forging capacity 3 - 4 times greater than present concepts.

2. Objective - The objective of this program was to scale-up the size and energy potential of high velocity, controlled energy (HERF) type forging machines to establish additional capacity for larger, net size, complex forgings required in modern air and space vehicles.

B. References

1. Standard Form 26, Award/Contract No. F33615-68-C-1578, Proj. No. TB-8-MMP-254/123-8, USAF, USAF Systems Cmd., Aeronautical Systems Div., Wright-Patterson AFB, Ohio, dated 22 April 1968, subject: "Award of Contract."

2. Technical Proposal, Contract No. F3361-68-R-1018, General Dynamics, Electro Dynamic Division, Avenel, New Jersey 07001, dated 6 September 1967, subject: "Design and Engineering Plans for 1-1/2 Million Foot Pound High Energy Rate Forging Machine."

3. Ltrs., Contract No. F33615-68-C-1578, Project No. TB-8-MMP-254/123-8, General Dynamics Corporation, Electro Dynamic Division, Avenel, New Jersey 07001, submitted periodically, subject: "Status Report No. 1 through 14."

II. BACKGROUND

In early development years, General Dynamics pioneered high energy rate equipment. In 1958, they introduced the

first DYNAPAK High Energy Rate Metalworking Machine (see paragraph I.B.2.). They have been eminent in this field of equipment research and increasingly active in the development of new and sophisticated methods of forging parts for the aerospace age. Prior to 1968, the maximum controlled energy potential of forging equipment was 550,000 ft./lbs. with a die space of 30" x 30" (see paragraph I.B.2.). Because of the increasing demand for greater capacity and more complex forgings, General Dynamics submitted a proposal (see paragraph I.B.2.) which resulted in the issuance of contract (see paragraph I.B.1.) requiring high energy type machines of the magnitude reflected in this report.

III. DESCRIPTION OF MATERIAL

The 1-1/2 Million Foot Pound Forging Machine, Model 2436, incorporates mechanical, hydraulic and electrical systems.

A. Mechanical System

The mechanical system is composed primarily of two (2) integral opposing rams weighing 73,000 pounds each, having a maximum impact velocity of 630 in./sec. which creates the rated energy of 1,500,000 ft./lbs. Only 1,500 psi gas ram pressure is required to attain this energy output.

B. Hydraulic System

The hydraulic system is composed of wo (2) 100 HP motors which actuate four (4) pumps (capacity 35 GPM) to provide fluid for a uniquely stored energy system capable of sustaining three (3) successive multiple blows of the rams in several seconds. All functions are interlocked to provide maximum smoothness of hydraulic fluid delivery with minimum inertia shock.

C. Electrical System

The electrical system consists of conventional electrical circuitry monitored and controlled by a single console desk.

D. Physical Characteristics

(See Appendix I)

IV. SYNOPSIS

A. General

1. The requirement for this design and engineering study was established by paragraph I.B.1.

2. This requirement was the outgrowth of a study submitted by General Dynamics on 6 September 1968 (see paragraph I.B.2.).

3. Periodic progress reports were submitted under separate cover (see paragraph I.B.3.).

4. A summary of computer programs and stress analysis is contained in Appendix III.

B. Mechanical System

1. The two (2) DYNAPAK ram assemblies, complete with separate gas chambers is the central feature which enables the equipment to function as a pure counter-blow machine. The two rams have been proportioned for extreme stiffness in order to support large tooling requirements. The proposed design results in the greatest rigidity possible, consistent with adequate guiding and minimum weight.

2. Complete tooling interchangeability between the upper and lower rams is provided. This is a significant feature since many tools, particularly those used for forging large parts, require as much tooling area for the upper tools as is required for the lower tools (92" x 75" on each ram). It is also significant that tooling can be interchanged after starting an operation in order to obtain particular advantages involving part removal, forming and ejection that cannot otherwise be predetermined.

3. Detailed stress analysis was performed on machine areas of high-stress concentration which resulted in the adoption of the following:

a. A structural frame with solid one piece slab side supports.

b. Integral one-piece ram-columns having adequate section modules to reduce point loading when subjected to eccentric loading. (See Appendix III, Plate III, "Bending Considerations and Side Load Capability.") This type forging results in strength, simplicity and reliability.

4. There is a minimum inertia loading of jack cylinders because the jacks are not subjected to eccentric forces since the loading is in the center of each jack cylinder. Therefore, an additional carriage with separate guidance and an arresting system is not required, greatly simplifying maintenance of the machine.

5. The jack recocking system is particularly advantageous for quickly separating the two rams after firing. No massive mechanical pivoting locks with their actuating gear and hydraulic controls are required. Consequently, the system is in the ready position within a fraction of a second, providing the multiple high capability.

6. A single fire control valve is provided for simultaneous firing of both rams.

7. Appendix I contains further details and illustrations of the machine.

C. Hydraulic System

1. The crux of the high performance accumulator system centers around a large common receiver (500 gal. capacity) feeding three (3) identical accumulators with a displacement of 75 gallons each. This assures almost negligible pressure drop in the accumulators.

2. The pressure differential created by a system of this capacity produces enough high pressure fluid for almost immediate retraction of the four 16-inch diameter jacks after impact.

3. The fractional time lag in piston operation and the utilization of a combination of the three accumulators provides the multiple hit capability with a minimum loss of heat to the forging.

4. The charging of the hydraulic system is accomplished by four (4) identical piston pumps, each having a rated pumping capacity of 35 GPM. Three (3) of these pumps are utilized to charge the accumulator bank and one (1)

pump supplies the fluid to both the ejector block and counterbalance systems.

5. A detailed discussion of the high performance accumulator system is contained in Appendix II.

D. Electrical System

1. The electrical system is controlled and monitored by an operator utilizing an Electrical Operating Console and a Control Stand.

2. The automatic events which are under control of the operator are as follows:

a. Main Automatic sequence.

b. Jack rotary locks.

c. Prefill or jack rapid advance cycle.

d. Jack retraction cycle.

3. Each of the automatic events is sequenced and functioned by a separate stepping relay, which insures that all programmed events in each category take place chronologically.

4. The stepping relay is so designed that if a malfunction occurs, such as a valve opening out of sequence, the integral permission light would not advance to its next successive position on command.

5. The system also contains a ready light which indicates that all safety features and initial conditions have been satisfied before the system becomes functional.

6. If the stepper fails to advance on command, or the system fails to function, the operator can view the last numbered position of the stepping relay through a transparent panel on the control stand or console and take appropriate action for subsequent trouble-shooting.

7. A detailed discussion of the electrical control system is contained in Appendix IV. Line items not specifically referred to are contingency steps resulting in the function of a major component.

E. <u>Miscellaneous</u>

1. Liaison was established with Lindberg Hevi-Duty, a Division of Sola Basic Industries, Rahway, New Jersey, to select an electric furnace which would be compatible with the overall design of the machine and which would satisfy all requirements as dictated by the intended usage of the machine. (See Appendix V.)

2. As a result of this liaison, it was determined that two (2) rotary electric furnaces would be suitable for installation on the foundation of the machine in its proposed configuration. (See Appendix V, Drawings D66-10 and VD-181.)

3. Each rotary furnace is 20'-0" in diameter and has a hearth width of approximately 3'-2". The heating capacity of each furnace is 5000 pounds per hour.

4. It is feasible to consider that one furnace could be used for pre-heating, thereby offering considerable flexibility in preparing unspecified materials for forging.

5. By utilizing both furnaces in conjunction with each other would result in a greater hourly output capability, at the same time retaining the potential of reheating forgings if temperatures should decrease excessively.

V. <u>CONCLUSIONS</u>

It is concluded that:

A. This design proposal provides a total controlled energy output in the magnitude of 1,500,000 foot pounds, with the capability of utilizing this energy in multiple blows.

B. This high energy potential provides the machine with the capacity of accepting larger and more complex forgings.

C. The multiple-impact principle provides high energy output transfer into a workpiece over a negligible time period, resulting in minimum loss of heat to the forging.

D. Multiple blows at varying energy levels will reduce the possibility of forging tool failures.

E. Controlled energy output will permit an impact energy capability consistent with the requirements for varied type forgings.

F. The counter-blow principle provides versatility in tooling capabilities.

G. The composite design of the machine emphasizes simplicity, flexibility, safety, minimum maintenance requirements and overall increased production.

APPENDIX I

Physical Characteristics

SPECIFICATIONS AND RATING

Energy	1-1/2 Million Foot Pounds at 1500 psi Gas Ram Pressure (Upper and Lower Ram)
Upper Ram Weight	73,000 Pounds
Lower Ram Weight	73,000 Pounds
Maximum Velocity at Rated Energy	630 Inches per Second
Die Area	92 Inches x 75 Inches
Daylight	7 Feet, 0 Inches
Stroke (Total)	3 Feet, 0 Inches
Overall Height	29 Feet, 4 Inches
Overall Width	18 Feet, 1 Inch
Overall Depth	9 Feet, 2 Inches

GENERAL DESCRIPTION

The 1-1/2 Million Foot-Pound High Energy Rate Forging Press, Model 2436, which has been designed under this contract is a pneumatic-mechanical type press of true counter-blow design.

MAIN RAMS

The mechanical system is composed primarily of two (2) integral opposing rams weighing 73,000 pounds each. The two (2) rams are controlled by individual pneumatic actuators which are capable of driving the rams at a maximum impact velocity of 630 in./sec. which creates the rated energy of 1,500,000 foot pounds. A sketch of the opposed ram arrangement is shown on Plate I.

It was necessary to design these rams so that adequate section back-up would be furnished to the large die dimensions which were specified with respect to stress and deflection of the main ram members. The section modulus is adequate to sustain loading substantially off the centerline of the press.

Results of our early computer studies confirmed our preliminary computations that the original specification velocity of 800 in./sec. could not be obtained with a conservative design or without certain unfavorable compromises.

The most significant part of this design is that both upper and lower ram bolsters are identical. Equal tooling areas in the two bolsters makes a larger class of work possible, particularly in coining where the upper and lower dies may be almost equal in area. The complete tooling interchangeability between the upper and lower rams is advantageous also in obtaining particular advantages when a forging operation is being started. This could involve problems of forming, part removal and ejection that might not otherwise be predetermined.

At an early stage of our design effort, we sourced primary forging vendors to determine if the proposed large ram forgings could be produced in one piece. The response was affirmative which resulted in further simplification of these important members.

The proposed design results in the greatest rigidity possible, consistent with adequate guiding and minimum weight.

EJECTORS

Both rams are equipped with hydraulic ejectors capable of 60 ton knock-out force with a maximum 8 inch stroke.

POWER SOURCE AND TRIGGERING SYSTEM

The machine is controlled by our patented triggering and sealing system. The power source to propel the rams is the Dynapak basic actuator. This is the same basic power source used on production model Dynapak machines.

The energy stored in compressed gas is released through a quick opening valve to propel the rams together at high velocity. Individual, but identical fire chambers are provided for each ram.

The fire chamber and seal base concept is shown in the drawing on Plate II.

The heart of any high energy rate machine is the valve used to release the energy of the stored compressed gas to act on the two opposed rams.

VALVE SEQUENCE

The Dynapak valving technique is extremely simple and production proven. It is illustrated, in sequence, for one ram only on Plate III.

1. The external hydraulic recocking jacks are engaged and actuated, moving one ram up and the other down. The residual driving pressure in the gas chambers is overcome causing the gas to assume its original volume and pressure in each chamber.

2. The valves are cocked when the ram heads seat against the main seal base. The original amount of gas required to trigger the system is trapped above the ram caps inside the main seals.

3. The small amount of trigger gas trapped above and below the two ram caps is vented to atmosphere. The external rotary jacks are then rotated 90 degrees and the machine is in static balance ready to perform a forging blow.

4. A small volume of high pressure gas is admitted into the areas inside the seal base ring. The slight downward movement of the rams away from the main seals allows the gas pressure in the chambers to act instantaneously over the entire area of the ram column. One ram is driven downward and the other upward by the equal and opposite gas forces. Each ram is acted upon with equal thrust and momentum, but in opposite directions.

When the two rams meet in mid-stroke, the useful work is accomplished in the die area. All forces generated are absorbed in the work piece or are cancelled out leaving the machine free of stress and vibration.

To provide for the next forging cycle, the external hydraulic jacks (4) have followed the two rams as they were being driven together. Upon engagement with the rams, they rotate 90 degrees and make locking engagement with the rams.

The hydraulic and accumulator system described in Appendix C now rapidly separates the two rams, allowing for ejection of the work piece and preparation for the next cycle. The recocking of the rams by the external hydraulic jacks has again inititated the valve sequence described in paragraph (1).

A single fire control valve is mounted equidistant from each high pressure gas chamber and common fire pressure is utilized in both chambers. This eliminates the need for a mechanical back-up triggering system. Accidental misfiring of one ram only is eliminated since when one ram fires, full fire pressure is then free to act on the opposite ram, firing it as a reaction.

The machine can be operated from <u>either</u> compressed nitrogen or high pressure compressed air. There are no hazards in using compressed air with the Dynapak actuator. Since the volume of gas required is rather large, economics would dictate the use of compressed air. We have included an adequate high pressure air compressor system and receiver in our design.

EXTERNAL HYDRAULIC JACKS

The rotating jack assemblies are shown in the drawing on Plate IV. By utilizing this unique external recocking system, we were able to simplify the machine in several respects.

1. Internal jacks would have had to be incorporated in the lower bolster which would have interfered with the effective tooling area.

2. Locating the hydraulic jacks on the top and bottom of the press removes them from the damaging area of forging scale and heat.

3. There is minimum inertia loading of the jack cylinders. The jacks are not subjected to eccentric forces since the loading is in the center of each jack cylinder. Therefore, an additional carriage with separate guidance and an arresting system is not required, greatly simplifying maintenance of the machine.

4. The system is advantageous for rapid separation of the two rams after a forging blow.

5. Note that double back-up seals were designed into this system to minimize down time due to oil leakage. Oil leakage has plagued many previous high pressure hydraulic installations, a fact which justifies the back-up system.

MACHINE PROPER

Plate V shows an artist's sketch of the assembled machine.

Plate VI shows a cutaway view of the assembled machine.

Plate XVII is a cutaway view of the machine with key features identified to the stress analysis report. (This plate will be found in Appendix C.)

Examination of these layouts as well as the stress analysis details will show that the outer structure design is adequate to contain moderately severe eccentric loading forces. The counter-blow system does not introduce any inertia loading into the outer structure.

The structural frame incorporates solid one piece slab side supports. These solid slabs provide support for the adjustable gib system.

Note that full length gibs are employed and that both rams are fully contained by the gibs throughout their entire strokes.

Further design details on the support structure and the gibs is contained in Appendix C.

MACHINE OUTPUT

The machine is rateo conservatively as far as developed energy is concerned.

The 1,500,000 foot pound specification energy is developed with only 1500 psi Gas Ram Pressure.

The machine has reserve capacity; however, since the fire chambers and the triggering system are designed for 2000 psi Gas Pressure which has been the standard on all Dynapak production machines.

The final dynamic computer run (see Appendix III) indicates that with fire pressure set at the design limit of 2000 psi and taking into account a computer integrated allowance for U-cup seal friction during the adiabatic gas expansion in the two fire chambers, the developed energy will be slightly in excess of 2,000,000 foot pounds.

A Theoretical Energy Diagram, Drawing BG-10525 and a Theoretical Ram Velocity Diagram, Drawing BG-10526 are included for reference purposes.

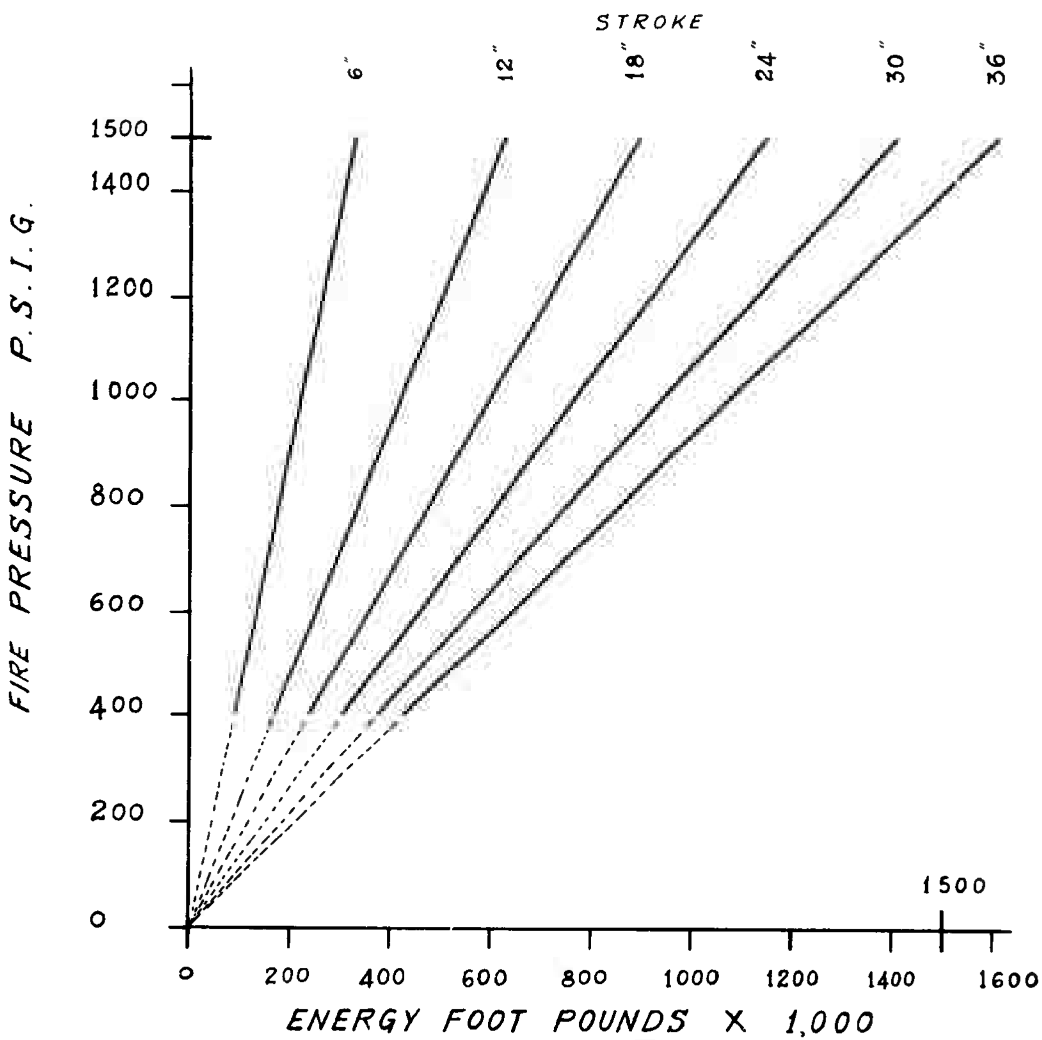


FIGURE 1

THEORETICAL ENERGY DIAGRAM

DYNAPAK

MODEL 2436

DWG. No. BG - 10525

BY : J. KOSKORIS

DATE 8 - 22 - 69

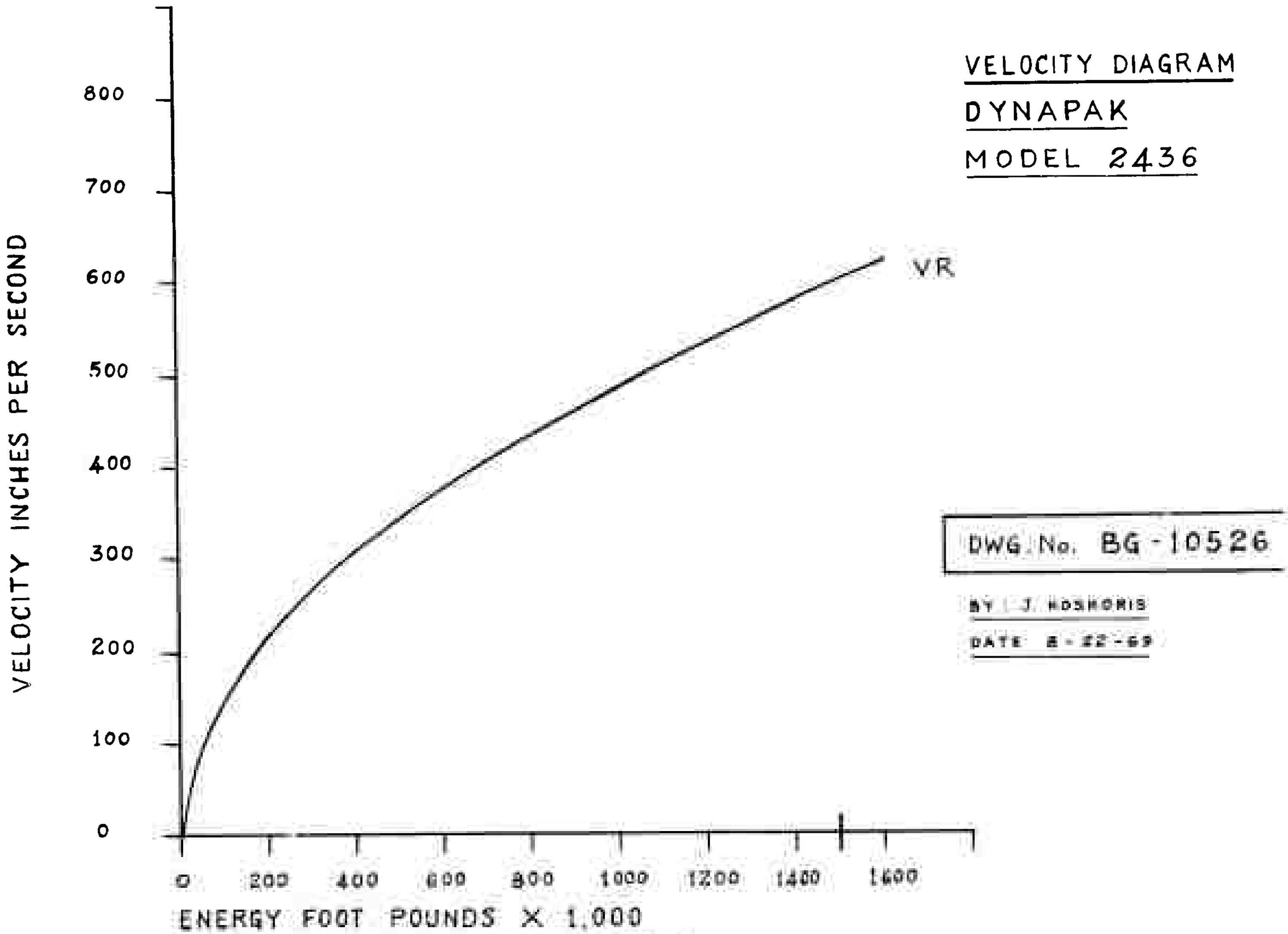
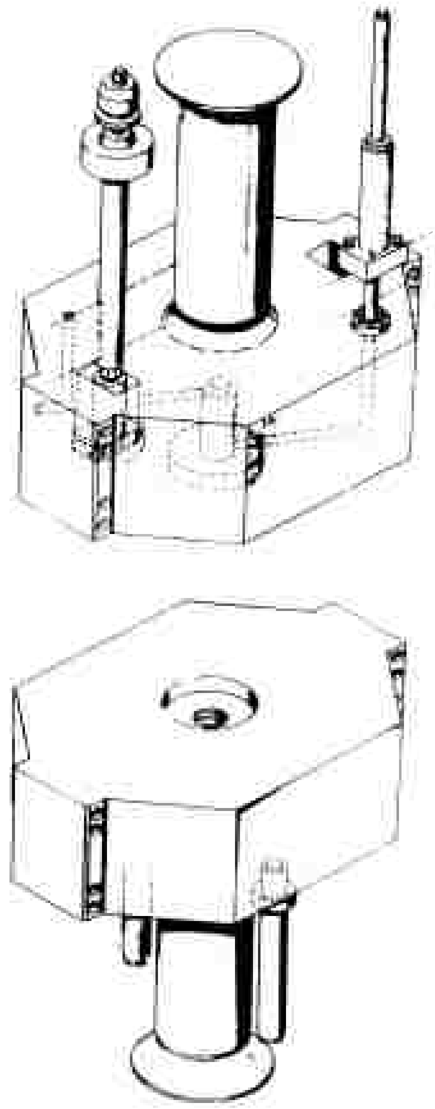
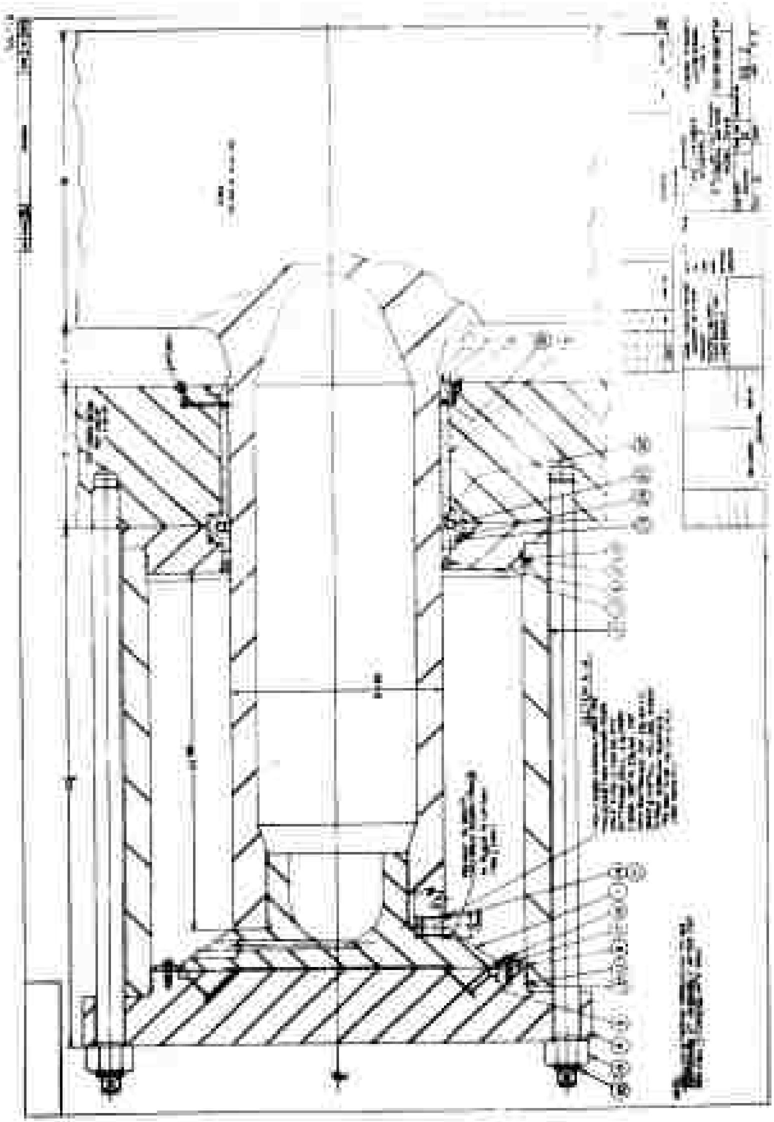

VELOCITY DIAGRAM
DYNAPAK
MODEL 2436
VR
VELOCITY INCHES PER SECOND
800
700
600
500
400
300
200
100
0
0
200
400
600
800
1000
1200
1400
1600
ENERGY FOOT POUNDS X 1,000
DWG. No. BG - 10526
BY I. J. KOSKORIS
DATE 8 - 22 - 69

FIGURE 2

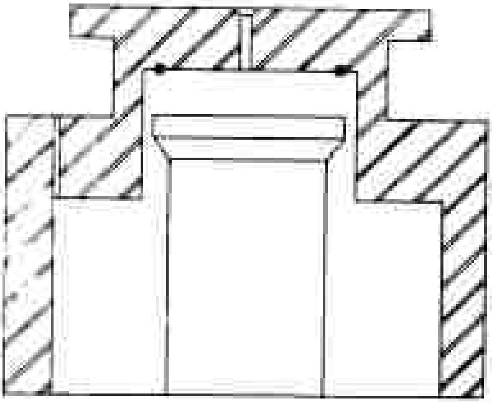


OPPOSED RAM ARRANGEMENT
PLATE 1

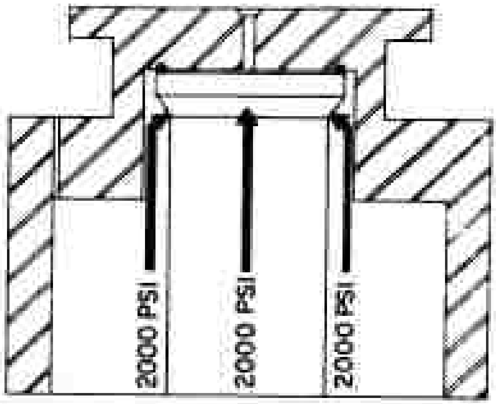


SEAL BALANCE CONCEPT
PLATE II

VALVE SEQUENCE



1



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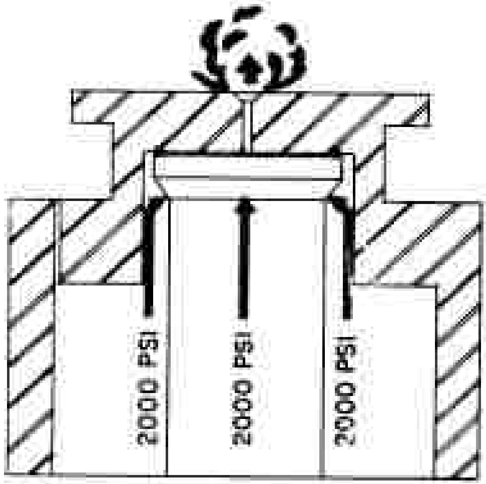

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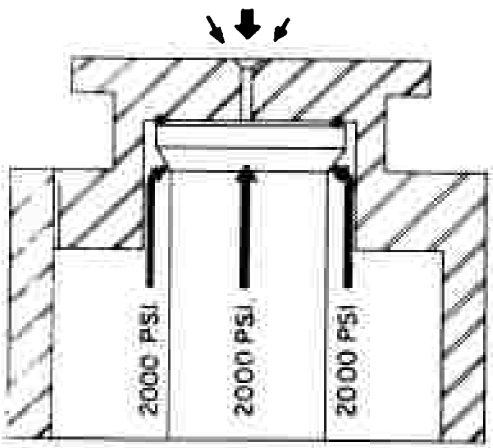

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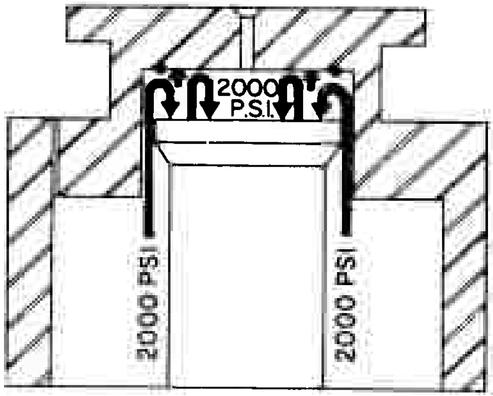
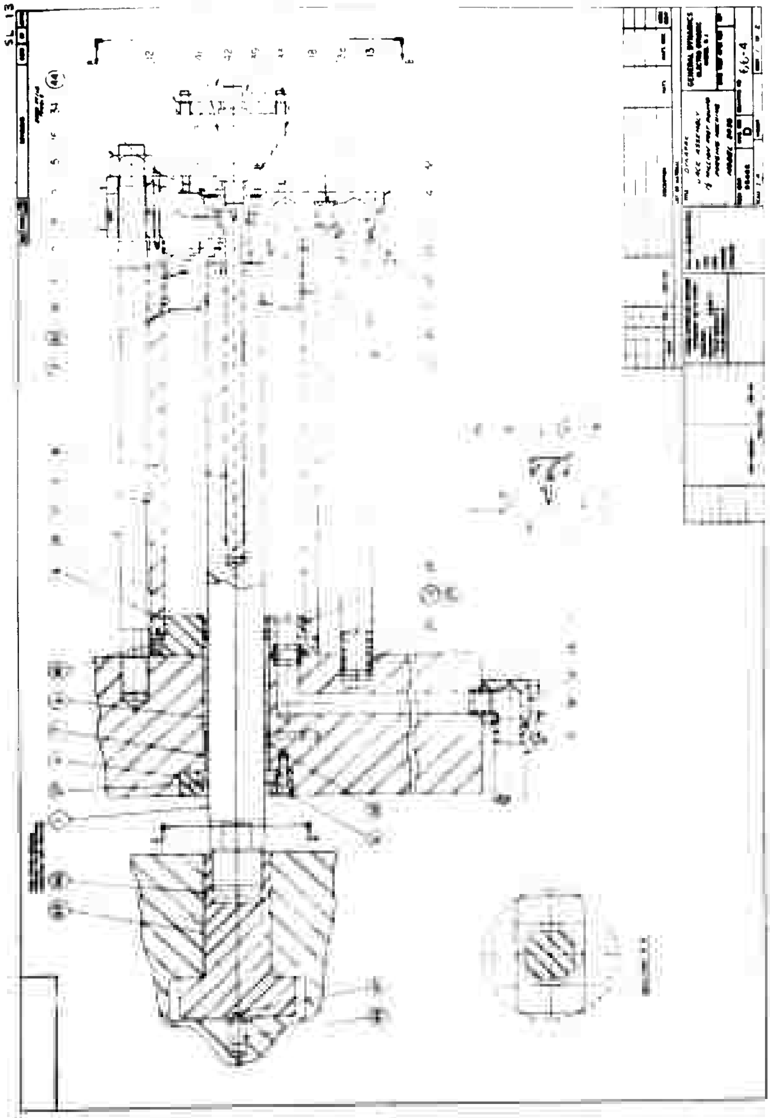
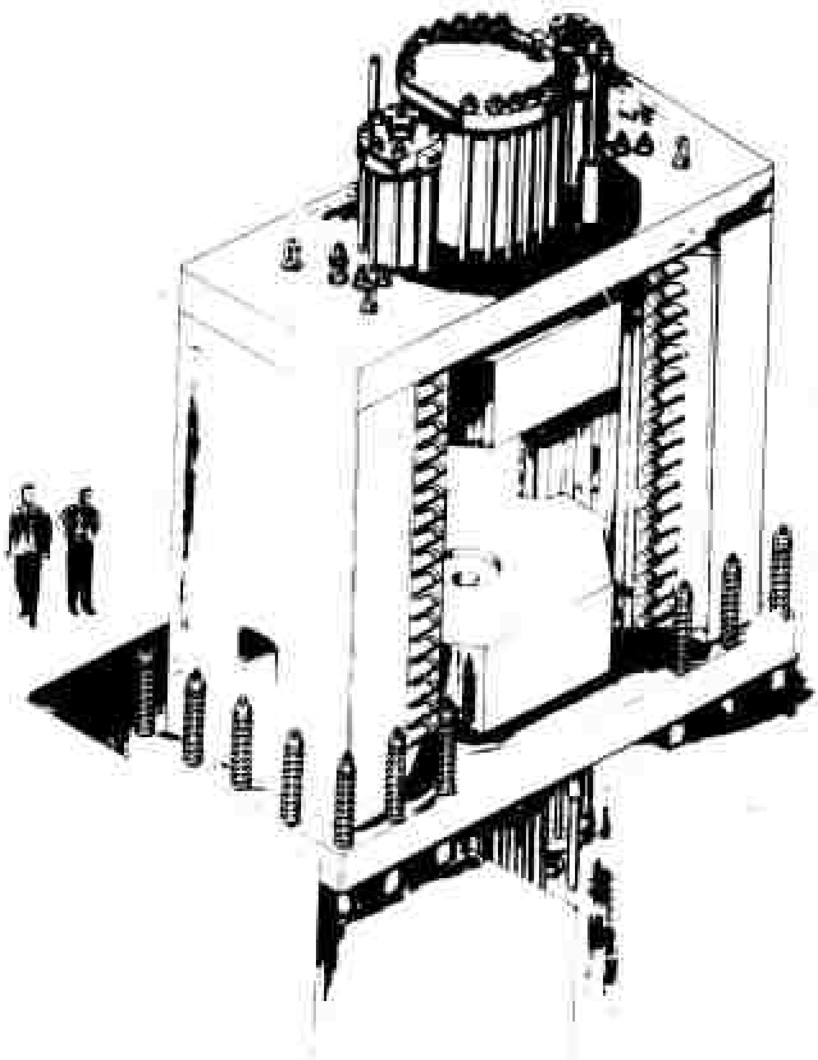


5

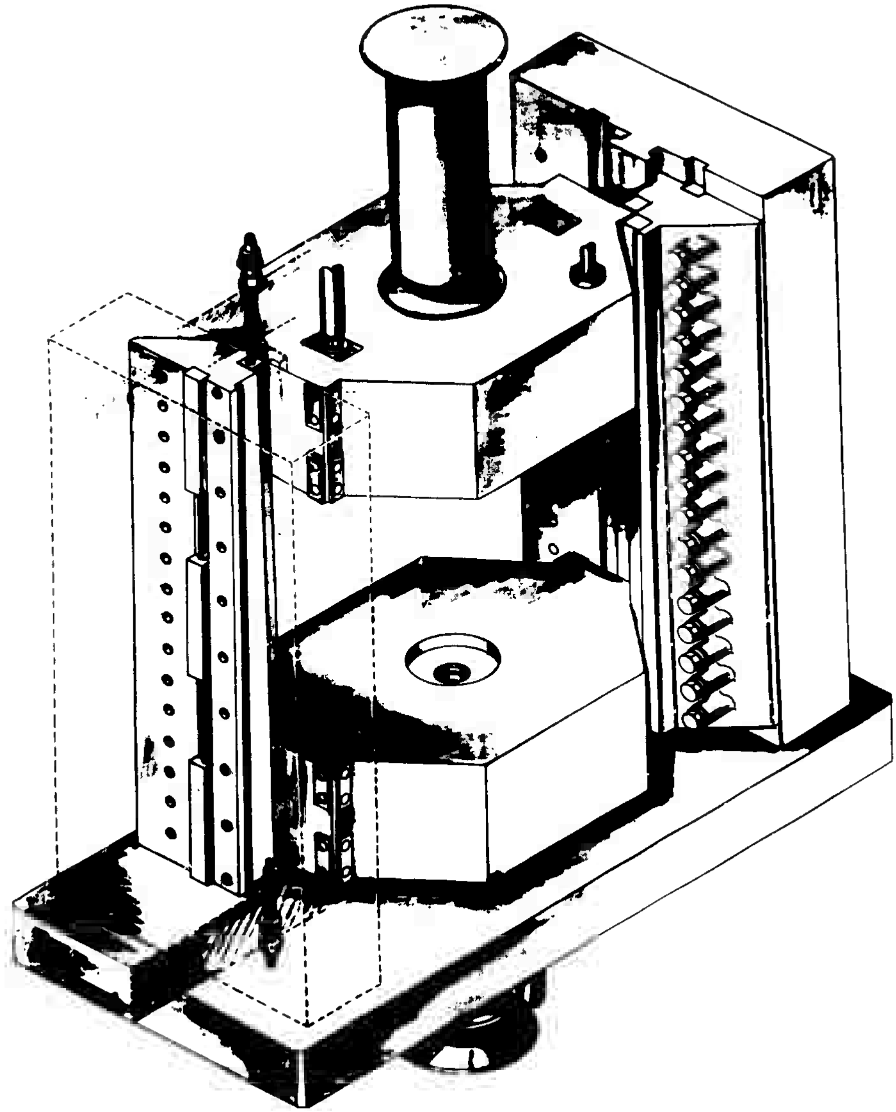
PLATE III



ROTATING JACK ASSEMBLY
PLATE IV



ARTIST'S DRAWING
OUTSIDE VIEW OF MACHINE ASSEMBLY
PLATE V



ARTIST'S DRAWING
CUTAWAY VIEW OF MACHINE ASSEMBLY
PLATE VI

APPENDIX II

High Performance Hydraulic and Accumulator System

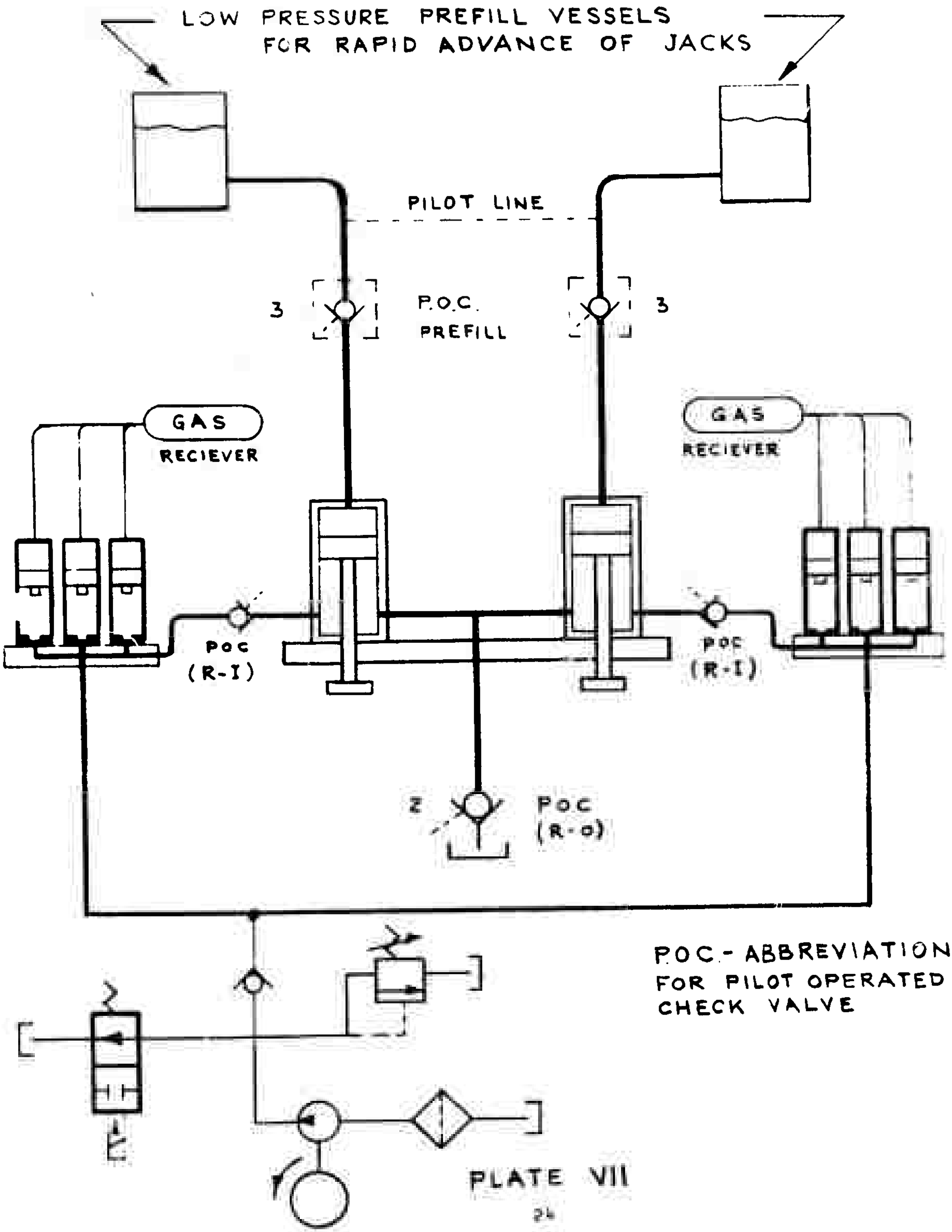
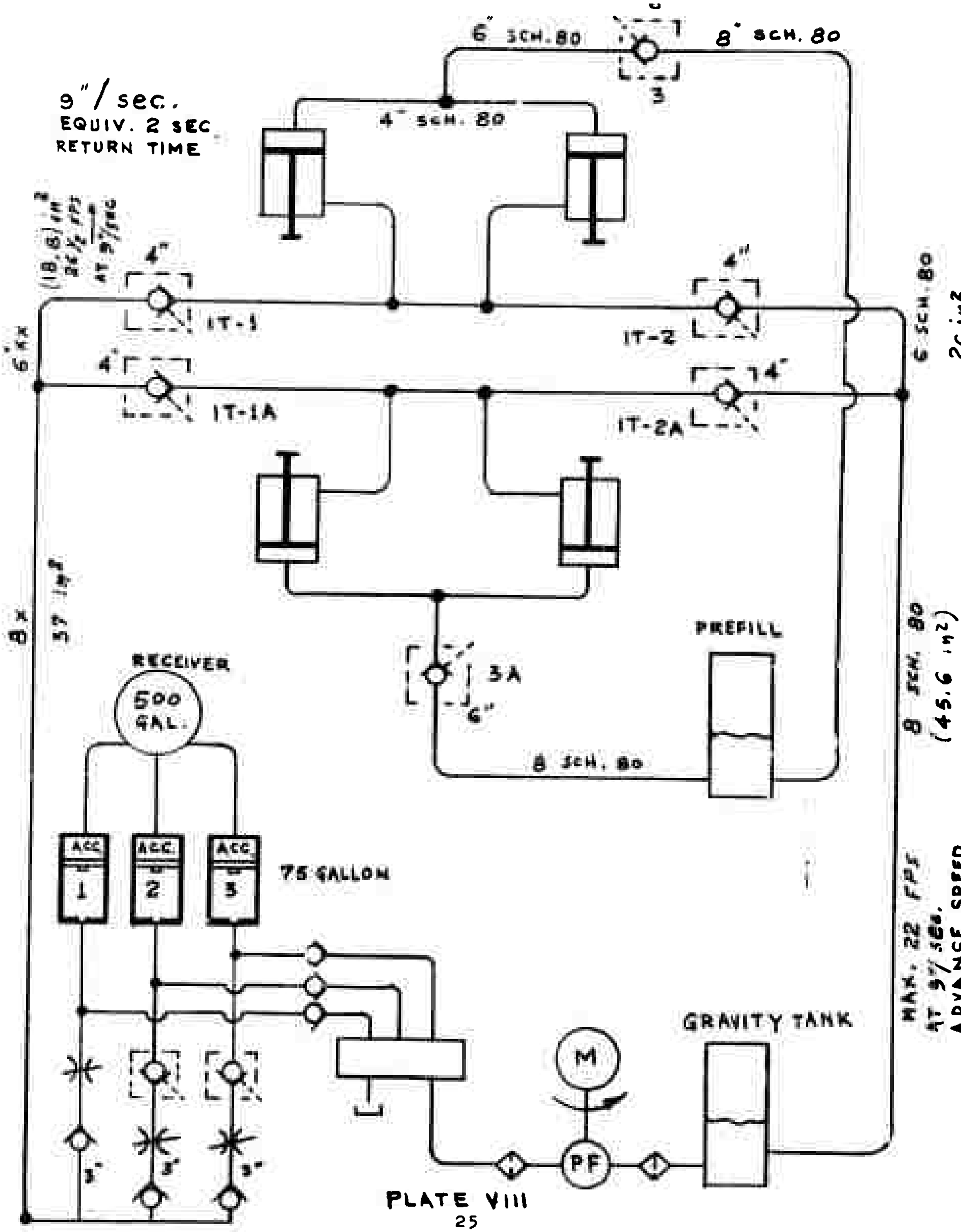

LOW PRESSURE PREFILL VESSELS
FOR RAPID ADVANCE OF JACKS
PILOT LINE
3
P.O.C.
PREFILL
3
GAS
RECIEVER
GAS
RECIEVER
POC
(R-I)
POC
(R-I)
2
POC
(R-O)
P.O.C. - ABBREVIATION
FOR PILOT OPERATED
CHECK VALVE

PLATE VII



6" SCH. 80
8" SCH. 80
3
9"/sec.
EQUIV. 2 SEC
RETURN TIME
4" SCH. 80
4"
IT-1
4"
IT-2
6 SCH. 80
4"
IT-1A
14"
IT-2A
PREFILL
3A
6"
RECEIVER
500 GAL.
8 SCH. 80
(45.6 in²)
ACC 1
ACC 2
ACC 3
75 GALLON
GRAVITY TANK
M
PF
3"
3"
3"

PLATE VIII

TYPICAL PILOT CIRCUIT WITH SEQUENTIAL PRESSURE SWITCH FEEDBACK SIGNALS TO MAIN CONTROL

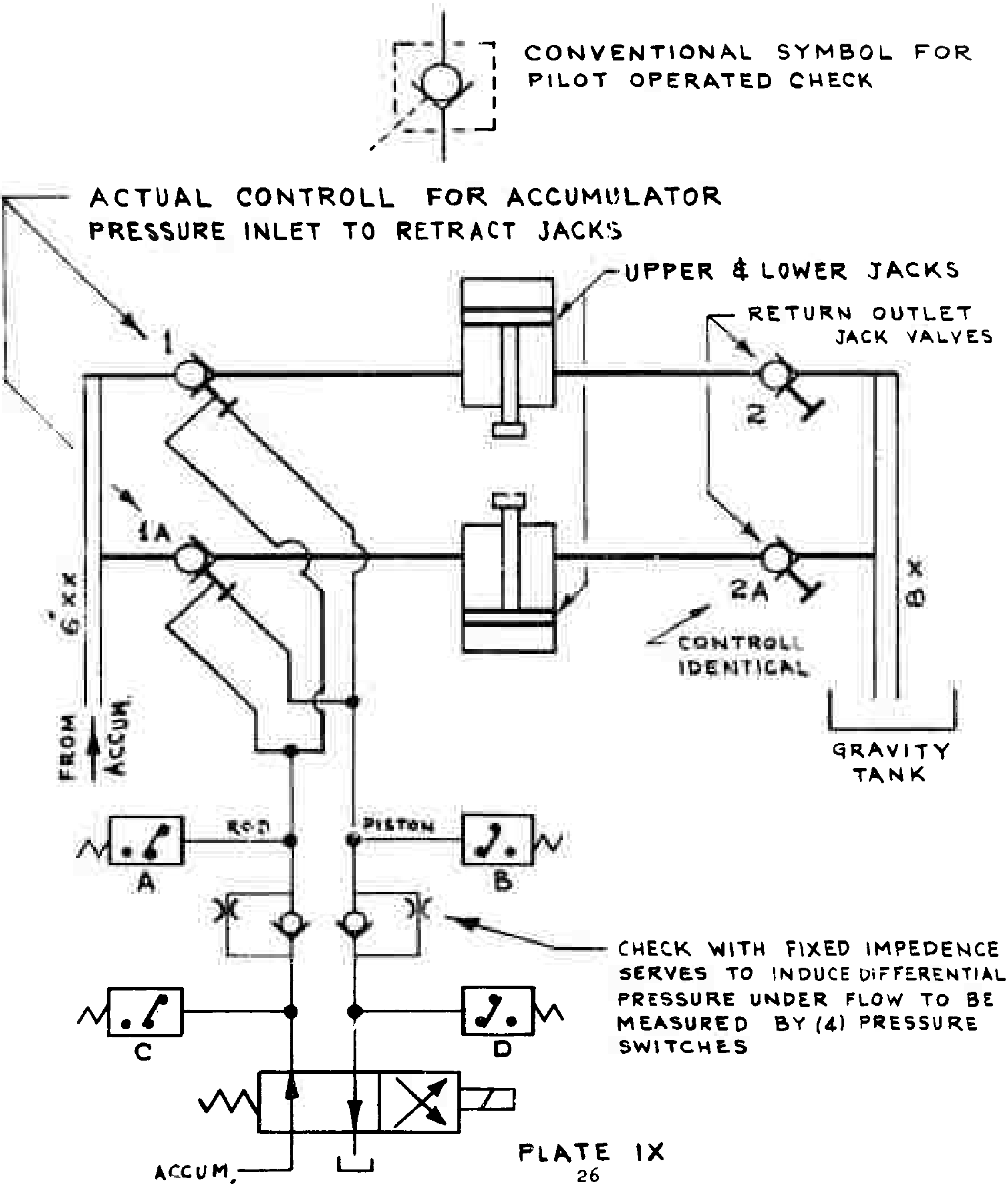


PLATE IX

ACCUMULATOR CONTROL

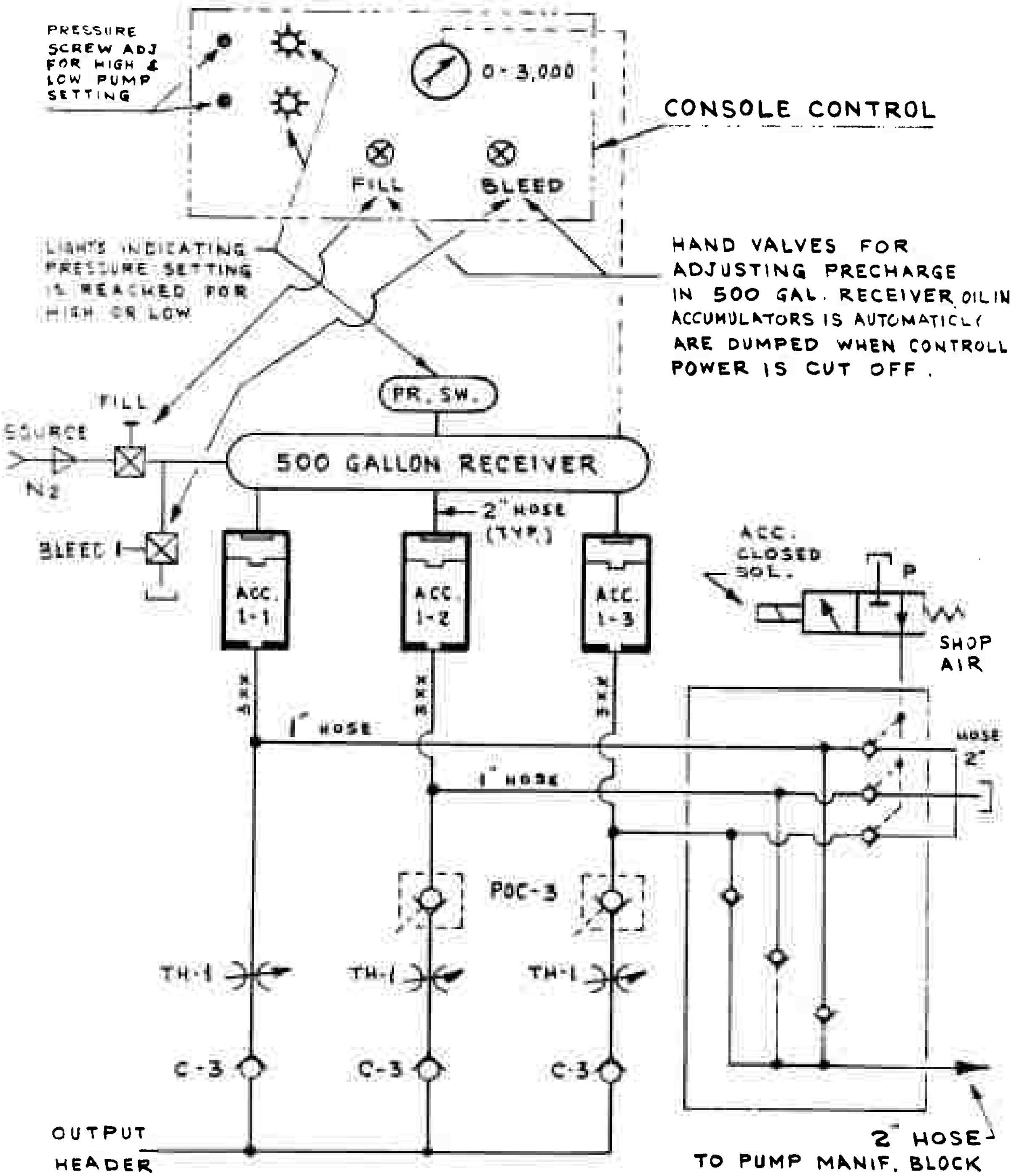


PLATE X

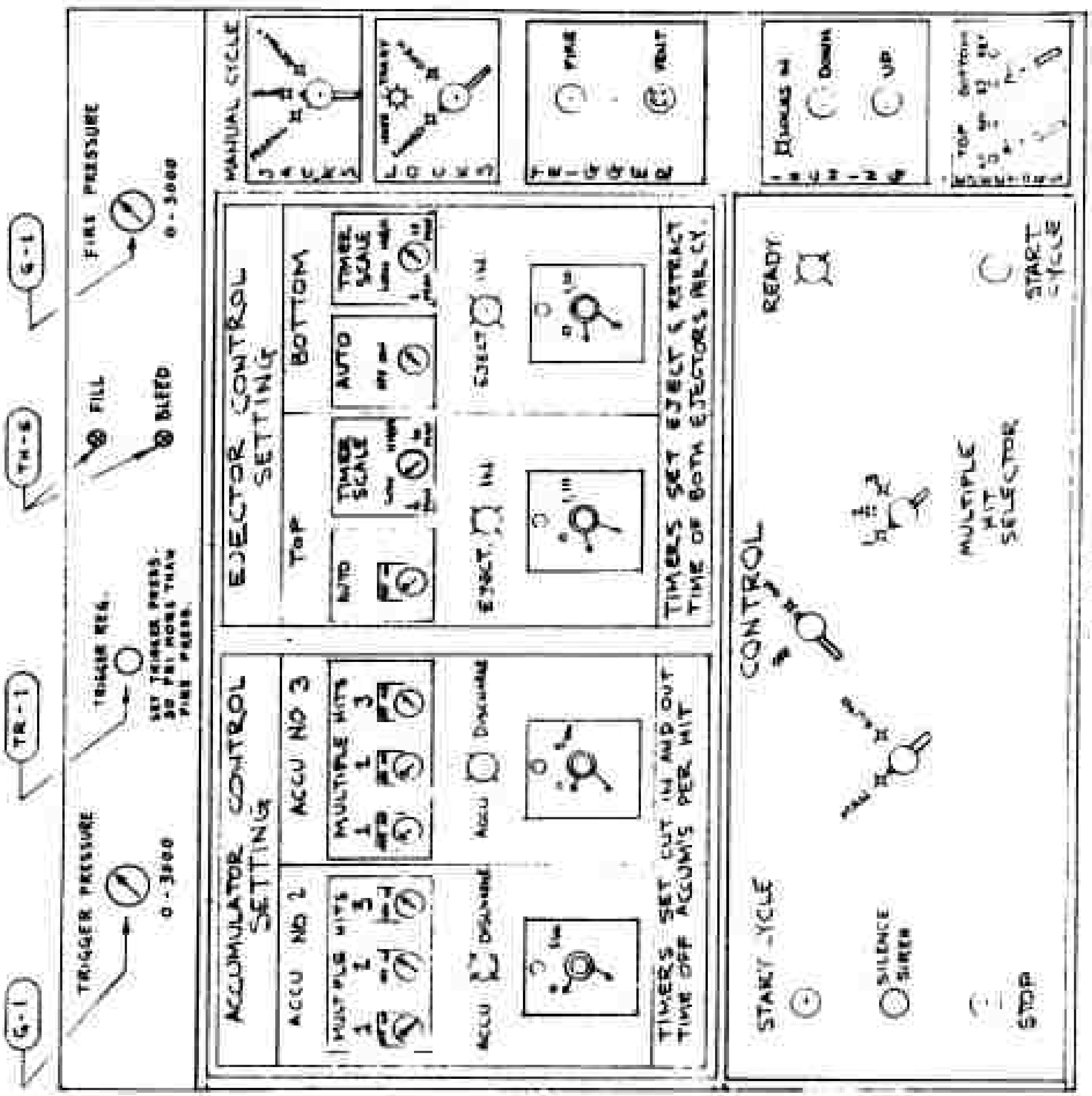

G-1
TR-1
TR-6
TRIGGER PRESSURE
0-3000
TRIGGER REG.
SET TRIGGER PRESS. 30 PSI MORE THAN FIRE PRESS.
FILL
BLEED
FIRE PRESSURE
0-3000
MANUAL CYCLE
FIRE
ACCUMULATOR CONTROL SETTING
ACCU NO 2
ACCU NO 3
MULTIPLE HITS
1
2
3
ACCU
DRAINING
TIMERS SET CUT IN AND OUT TIME OFF ACCUM'S PER HIT
EJECTOR CONTROL SETTING
TOP
BOTTOM
AUTO
TIMER SCALE
EJECT IN
TIMERS SET EJECT & RETRACT TIME OF BOTH EJECTORS IN. CY.
DOWN
UP
CONTROL
READY
START CYCLE
SILENCE SIREN
STOP
MULTIPLE HIT SELECTOR

PLATE XI

The hydraulic portion of a pneumatic high velocity machine restores the energy released by the machine.

This Hydraulic and Accumulator System is capable of furnishing approximately 4,500,000 foot-pounds of useful energy. This energy is to be delivered by means of a new concept over three successive blows of 1,500,000 foot-pounds.

Since the first use of HERF machines, ten years ago, there has been a trend to increase the energy delivered to the workpiece and to reduce the time required to complete a single cycle. Within the last six years, machine cycle time has been reduced from twenty seconds to an average of six seconds per cycle.

Equipment sold to date has had an energy output increase from 140,000 to 225,000 foot-pounds.

It was evident that this double demand for greater power with a faster delivery rate would increase both the size and importance of the role the hydraulic power unit plays in restoring energy to the high velocity machine.

In the design of the 1,500,000 foot-pounds machine, top priority was given to making the hydraulic system powerful enough to have equivalent or better cycle time than present machines.

In addition, the desirability of multiple, quick succession blows which would allow the energy delivery to be doubled or tripled in a short period of time has been desired by many users. It was decided that this capability would be highly desirable on the 1,500,000 foot pound machine.

To achieve this high performance without unusually large plumbing and a large number of pumps and motors, it became obvious that a large accumulator stored energy system would have to be incorporated in the design.

This decision to employ a modern, large capacity, piston accumulator system enabled us to utilize standard proven motor and pump components for the pumping station. Two (2) 100 HP, double shaft motors drive four (4) Dynex, 35 G.P.M., piston pumps to provide fluid for the accumulator system. This pump and motor combination has been field proven on our production model 1220D Dynapak machines.

We have concentrated considerable effort over the past several years on the improvement of the hydraulic system on our standard line of Dynapak machines, and these field tested improvements have been incorporated in this design.

All valves are sub-plate mounted to manifold blocks to reduce the number of pipe joints and the usual associated problem of leakage.

All valves and lines are adequately sized and all functions are interlocked to provide maximum smoothness of hydraulic fluid delivery with minimum inertia shock.

The system has been designed for petroleum-based hydraulic fluid. Our own experience with the fire-resistant fluids, particularly the phosphate-ester types, has convinced us that they are a primary source of field maintenance problems. Also, their poor lubricating qualities and low viscosity index, cause wear on the finely machined parts of sophisticated systems.

The first concept for the accumulator system is shown in the hydraulic schematic on Plate VII. Advantages which would result from this first design approach are as follows:

1. A minimum of connecting piping would assure practically no pressure drop between high and low pressure stored systems. (To reduce prefill plumbing to a minimum and get perfect symmetry, a separate prefill vessel for each jack cylinder was planned.)

2. An unusually symmetrical, almost pipeless, high pressure accumulator piping arrangement was a key feature. Unfortunately, several disadvantages of the first concept became apparent. They are:

    1. Identical systems for top and bottom jacks would be required. Therefore, two sets of hydraulic pumping equipment would be required, giving rise to unnecessary complexity.

2. The multitude of pressure vessels required considerable space both above and below the floor and considerably reduced access to mechanical members at the top of machine.

3. The arrangement would have required six accumulators protruding above the floor in the same relative location as the upper set and this would reduce access to the machine at floor level.

4. It was difficult to determine a location for the gas receiver bottle on the lower set.

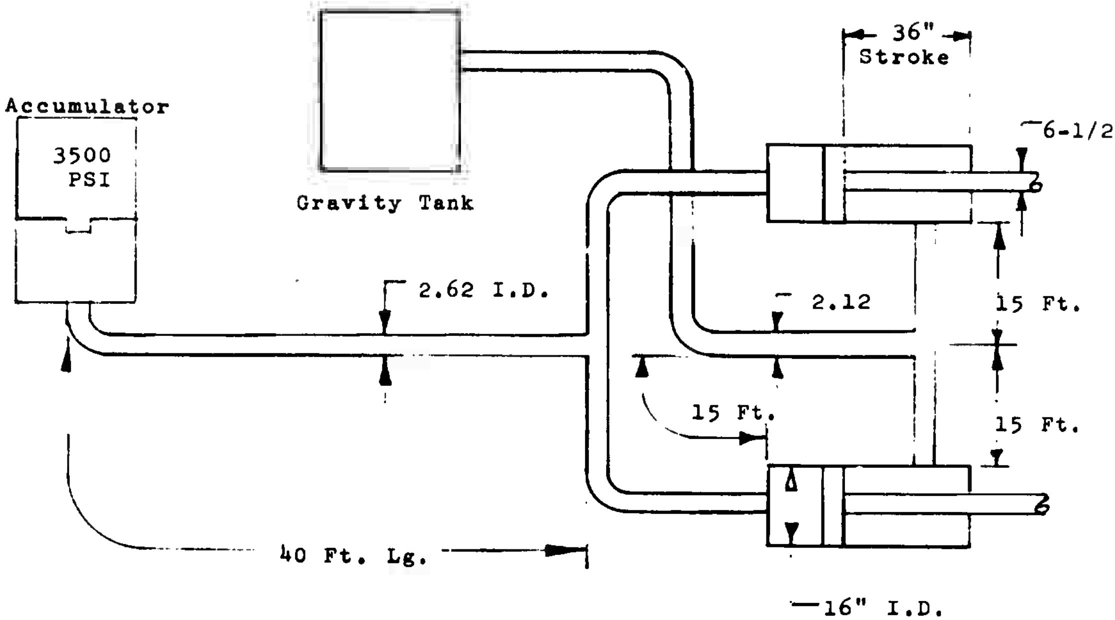
5. Heavy hydraulic equipment on the top of a forging machine gives an unfavorable weight distribution to the machine assembly and has disadvantages in maintenance when leaks can drip down to the forge shop. Access by means of high ladder is less attractive than descending a stairway to a hydraulic room located below the floor out of the forge shop environment.

   Safety considerations in having the tanks and pumping station out of the furnace and forge environment are obvious.

COMPUTER STUDY OF ACCUMULATOR SYSTEM

A complete study of a branch pipe system was made to study the effect of pipe length and friction on system power losses for any particular system studied. From this study, it was found that allowing three seconds for retracting both jack systems 36 inches, caused a pressure drop and the resultant horsepower losses were very low. The plumbing system model of the computer study is shown on Plate XII. The hydraulic accumulator system study is discussed in Appendix III.

PLATE NO. XII



Allowing two seconds for full retraction produces a loss of 215 horsepower and pressure drop of 235 psi for the above piping network. Considering that the main work power is 1917 at this velocity, an overall efficiency of power delivery will still be an attractive $\frac{1917}{1917 + 215} \times 100 + 90\%$. The final piping network will have a lower resistance since it will utilize an 8XX and 6XX header from the accumulators to the rod connections of the jacks. The inside diameter of the 6XX pipe would be 4.9 inches and would carry 587 GPM at one foot per second. Consider the [illegible]er jack set taking the computer value of 1568 GPM. The maximum velocity through the 6XX pipe would only be $\frac{1568}{58.7} = 26 - 1/2$ FPS. At this velocity, pressure drop and power loss should be neglible for the lengths involved. Note that such large pipe sizes allow

the four 16 inch diameter jacks to retract at an average rate of nine inches per second. At this rate, they will be delivering 1917 horsepower (as previously discussed) to the gas ram systems and will be accepting 1568 GPM from the accumulating system, which will only produce a 26 - 1/2 FPS velocity through the 6XX supply header pipe.

SUMMARIZING PRINCIPAL ADVANTAGES OF FINAL SYSTEM SHOWN IN PLATE VIII COMPARED TO PLATE VII

1. Minimum redundancy - Single vessels for the prefill and gravity sump tank are located in a separate room below the floor.

2. Accumulator - One set of three accumulators located in the hydraulic room below the floor replaces twelve accumulators originally located on the side of the machine.

3. Accumulator Drive - The need for two pumping systems is eliminated with a common accumulator drive for upper and lower jack cylinders.

4. Minimized Water Hammer - Minimum inertia effect is retained by still mounting the main control valves close, (within six feet) to cylinder control ports.

MULTIPLE HIT CAPABILITY

The hydraulic diagram, Plate VIII, shows that the three 75 gallon piston accumulators are the heart of the design. They store enough high pressure fluid to immediately retract the four, 16-inch diameter jacks after impact.

Due to the large capacity of these accumulators and the need to keep piston velocities reasonable, it was decided to design special accumulators having the following features:

1. Use a 15-inch diameter bore to keep total package length more manageable.

2. Utilize a special 1/2-inch diameter urethane o-ring as a piston seal.

3. Utilize wear rings on either side of main seal.

4. Porous chrome-plated bore retains lube and reduces seal friction. Items 2, 3 and 4 are based on our successful experience with high velocity actuators.

5. Design utilizes 2-inch tie bolts rather than threaded cylinder. (Large threads tend to be troublesome on disassembly.)

6. Incorporate piston dashpot on fluid end to eliminate mechanical shock on bottoming.

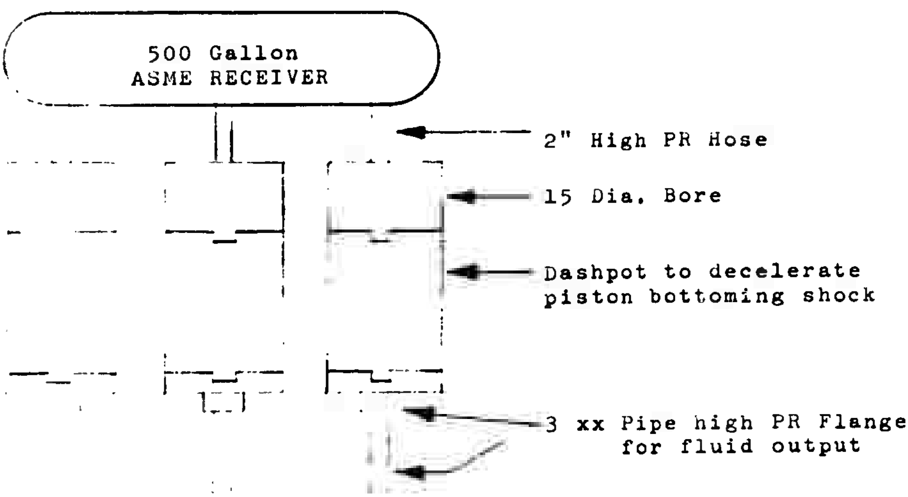
7. The gas heads of the three accumulators have 2-inch NPT connections permitting their connection to a 500 gallon receiver that will assume full usage of the stored fluid in each accumulator without excessive pressure drop.

CAPABILITY OF STORED ACCUMULATOR POWER

Consider the accumulator arrangement in its ready condition, Plate VIII. Note that all three accumulators have a common large receiver assuring them of almost negligible pressure drop for their displacement of 52 gallons.

PLATE XIII

EXAMPLE OF FLAT PRESSURE CHARACTERISTIC OF SYSTEM



Assume $P_1 = 3000$ PSI

For adiabatic expansion

$$P_1 V_1^{1.38} = P_2 V_2^{1.38}$$

$$\left(\frac{V_1}{V_2}\right)^{1.38} = \frac{P_2}{P_1}$$

$$\frac{P_2}{P_1} = \left(\frac{500}{558}\right)^{1.38} = (.895)^{1.38} = .86$$

Hence $P_2 = 3000 \times .86 =$ 2600 PSI

EASE OF CONTROL

To maintain maximum speed consistent with minimum shock of cylinder movement, both the gas precharge for the accumulators and oil fill pressure are separately adjustable, providing maximum flexibility. In this way, stored oil energy can be varied in almost a limitless number of combinations of stiffness of accumulator as compared to stiffness of gas ram proper.

ACCUMULATOR CHARGING CIRCUIT

Accumulator control on the operating console is shown on Plate X. A schematic diagram, illustrating the accumulator automatic drain to the tank when the control is shut off is also indicated on Plate X. This permits a reading of the common precharge to all three accumulators and simple adjustment from the console.

CYLINDER SHOCK CONTROL

By matching the precharge gas accumulator pressure to the gas pressure in the machine cylinders, the optimum choice of mechanical high speed operations of the jack cylinders consistent with minimum hydraulic shock to the system can easily be met.

SEPARATE ACCUMULATOR DISCHARGE CONTROL BY TIMERS

On Plate X note that the accumulator #1-1 is always connected to the output header. Its rate of delivery can be controlled by throttling valve TH-1. Consider the accumulator as a fixed output pump that is always

available to retract or clamp the jacks up at a velocity that would introduce negligible mechanical shock on bottoming. Its companion accumulators, labeled Acc. 1-2 and Acc. 1-3 act as short burst high performance devices that are capable of instantaneously separating the jack cylinders after forging. Continued movement at this high velocity would produce excessive mechanical strain on the machine components. Although each jack cylinder (Plate IV) is furnished with a dashpot orifice plug and an integral check valve to cushion retraction at high speed, the machine will perform smoother with a slow down speed before bottoming on the retraction stops.

For example, assume that only one hit will be made and it is desired to get maximum (almost instant) release of energy from all three accumulators. Refer to Plate XI and set the timers for Acc. 2 and 3 to go on after .1 and .2 second delays, to stagger the input shock pulse and go off at .6 and .8 second bursts respectively. This simple procedure will make the accumulators release energy in .5 second bursts staggered slightly, both going in and out to achieve uniform acceleration and deceleration of the ram systems. The throttling valve opening in each accumulator line can be adjusted to control the "accumulator burst velocity" to velocities between 6 and 24 inches per second.

TYPICAL SETTINGS OF CONTROLS FOR THREE HITS

The time setting of both accumulators, 2 and 3 above, may be left for .5 second total duration bursts slightly staggered so that they can both be released together on the last hit.

Set the controls on Plate XI as follows:

Repeat Hit Selector		Set To 3
Acc. #2	Hit 1	On
	Hit 2	Off
	Hit 3	On
Acc. #3	Hit 1	Off
	Hit 2	On
	Hit 3	On

The following discharge sequence will occur as the rams hit the closed die for three successive blows.

Hit 1 - (1) Acc. #1 - fixed opening of 6 inches per second

(2) Acc. #2 - will cut in .1 second later than Acc. #1 and stay in .5 second, boosting speed another 6 inches per second to 12 inches per second but will drop out before the jacks fully retract.

Hit 2 - (1) Same as 1.

(2) Acc. #3 will perform task as above in #2 except for an additional .1 second delay.

Hit 3 - (1) Same as 1.

(2) Both Acc. #2 and #3 will give the final blow retraction the highest speed, possible up to 30 inches per second and sequence the power bursts for minimum shock.

SIMPLE POWER RELEASE CONTROL

Large amounts of hydraulic power discharge are controlled by selector switch and solid state timers are easily pre-set by the console operator.

It is clear that by console choice of timing bursts and setting of accumulator throttling valves, almost infinite control of stored hydraulic power can be effected. For short bursts of energy release it is conceivable that flow rates from the three accumulators in excess of 3137 GPM, and developing useful work overcoming gas cylinder load in the main cylinders exceeding 3835 horsepower, will be accomplished with high efficiency. These figures are taken from the first line of the computer printout sheet and correspond to a jack velocity of 18 inches per second against an almost constant gas load of 1400 PSI in the main fire chambers. The equivalent hydraulic balance pressure in the jacks is 2094 PSI. Since the accumulator design capability has been set at 3500 PSI, if follows that considerable power is available in the accumulators for achieving any reasonable velocity desired.

CONTROL OF ACCUMULATOR POWER BY MEANS OF TWO SIMPLE FINGER TIP CONSOLE ADJUSTMENTS

1. Console operator can read common accumulator precharge by shutting off control and dumping stored oil back to gravity tank. Individual 1 inch pilot operated checks, connected to each accumulator, will release the stored power making the system safe for maintenance and permitting the precharge to be read on the console, Plate X. The operator can now add or subtract precharge nitrogen by means of a fill and bleed set of hand valves on the console. This control permits the operator to harness the power output of the accumulators in the same manner that the main energy release within the fire chambers is controlled.

2. The operator can also vary the fluid charging pressure by varying the console pressure switch and adjustment screw that bypasses the accumulator pumps when the desired high pressure is reached. He has a similar adjustment for the cut-in or low pressure point at which the pumps will begin to charge the accumulator. Console lights, adjustment screws, and accumulator pressure gauge make this setting simple and easily adjustable.

HIGH VELOCITY PLANISHING

The finger tip adjustments make matching the stored fluid power in the accumulator to the stored gas power in the pneumatic mechanical portion of this machine practical, thus permitting for the first time, the concept of high velocity planishing with multiple blows.

PREFILL VESSEL DESIGN

Stabilization of fluid level will be achieved by three ultra-sonic detectors located in the vessel wall. Excessive or insufficient fluid level will automatically cause a correction using accumulator fluid to add to or extract from the fluid in the gravity vessel.

SEQUENCE INTERLOCKING FOR MAIN CONTROL VALVES

All critical controls have interlocing, based on pressure switches that sense static and dynamic fluid flow. For example, consider the main return outlet valve, 1T-1 in Plate VIII. Both these valves are pilot operated checks and have counterparts for the lower cylinder system denoted 1T-1A and 1T-2A.

Plate IX indicates four distinct modes for any cylinder. All pilot operated checks used for main functions such as accumulator flow control utilize double acting cylinders for better control. Each of the above mentioned modes of flow are measured by pressure switches and remembered by relays, thus interlocking the circuit to ensure that a return to tank valve has closed before admitting accumulator fluid to the cylinder.

Plate IX gives a detailed study of the pilot controls.

To reduce effect of fluid inertia or water hammer we elected to place the main control valves, inlet 1 and 1A and outlet 2 and 2A, as close to cylinder ports as practical and control the respective inlet and outlet operating cylinders with single valves as shown in Plate IX.

A simpler arrangement would be to use a single larger valve to replace 1, 1A and 2, 2A. However, the much longer piping network necessitated by this approach would increase the water hammer effect.

In order to ensure proper sequential control, pressure switches placed about FIXED HYDRAULIC ONE WAY IMPEDANCES are used to signal the difference between dynamic and static conditions of the cylinders controlling the main pilot operated checks.

By means of a "TRUTH TABLE" based on the outputs of these pressure switches, proper sequential relay signals from these pressure switches can be used to signal a stepping relay to stop the machine cycle or go ahead to the step if conditions monitored by the pressure switches are OK. The "TRUTH TABLE" arrangement of the pressure switch contacts act as limit switches within the operating cylinders and flow switches to detect movement of the cylinders in a particular direction. Hence, after the control valve solenoid has been given a signal to "DO" a task, feedback from the task must also take place before driving the control stepper to its next event position. The operator at the console receives an indication of where the automatic is at any instant by means of a digital readout device on the upper right hand portion of the console desk, Plate XIV.

TRUTH TABLE LOGIC — An indication of "TRUTH TABLE" logic that can be stored by relays is indicated below for the pressure switches in Plate IX.

Note: All pressure switches are set at some pressure less than that created by the impedance of the fixed orifice under flow.

KEY--------X DENOTES CLOSED
O DENOTES OPEN
⊠ DENOTES UNRELIABLE - DO NOT MEASURE

PR SWITCH N.O. CONTACTS	A	B	C	D
RELAY 1 - CYCLE IS RETRACTED	X	O	X	O
RELAY 2 - CYCLE IS EXTENDED	X	⊠	O	X
RELAY 3 - CYCLE IS EXTENDED	O	X	O	X
RELAY 4 - CYCLE IS RETRACTING	⊠	X	X	O

110 V

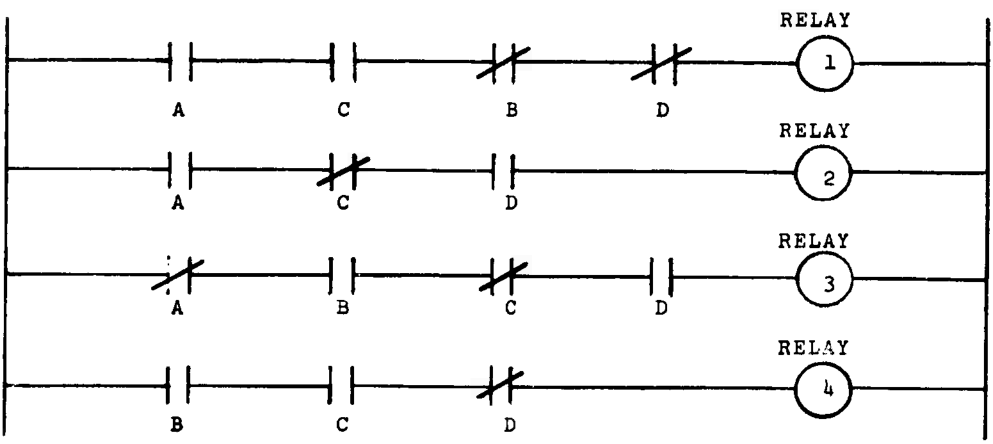


PLATE XIV

SUMMARIZING

Four modes can be extracted and so require the stepper to note feedback before going ahead.

APPENDIX III

Summary of Computer Programs and Stress Analysis

GENERAL

Our utilization of the IBM 1800 computer for design studies and for stress analysis studies is, we believe, unique for the press industry. It enabled us to include in our programs the effect of more variables than are usually accounted for when designing this type of equipment.

The prime advantage, of course, is the assurance that the final design is the most satisfactory possible from the alternatives. However, we feel that further refinement of the computer design technique would provide an accurate and rapid method of sizing other presses, both upward and downward from this particular design.

This summary of the various computer programs and their results is therefore included in this report.

This information plus copies of the actual computer print-outs was submitted as Data Item B003 under the contract, "Stress Analysis Report." For simplification, the actual computer print-outs are omitted from this report.

Program I - Study of Initial Velocity vs. Basic Parameters of Fire Pressure, Stroke, Net Impact Time and Energy

A simple program was written which would store design inputs of dimensional data, volumes, weights and varying gas pressures. The program then would yield outputs of impact velocity, exact point of collision, impact time, final reduced pressures due to adiabatic gas expansion and total net energy developed as a printed out quantity.

It would compute this data to a great accuracy since it made individual computations for units of time as small as 1/10,000 second, "DT" being an input quantity.

In this Fortran Plan, U-Cup friction and the simple cushion system parameters of diameter and pressure are taken into account along with allowance for tooling weights, "Toolwt."

No. 1 FORTRAN

```
      READ(2,1)VOL,URMWT,DRMWT,STR,TOOLW ,URFP,DRFP,DT,URDIA,DRDIA,SP,SD
    1 FORMAT(1F6.0,2F5.0,1F3.1,1F5.0,2F4.0,1F4.4,2F2.0,1F3.0,1F3.2)
      URMAS=(URMWT+.333*TOOLW )/386.
      DRMAS=(DRMWT+.666*TOOLW )/386.
    2 T=0
      URSTR=0
      DRSTR=0
      URVEL=0
      DRVEL=0
      URP=URFP
      DRP=DRFP
      DRA=.785*DRDIA**2.
      URA=.785*URDIA**2.0
    3 T=T+DT
      URTHR=URP*URA-URP*URDIA*.2+.333*TOOLW +URMWT
      SETFO  =(.785*SD**2.)*SP
      DRTHR=DRP*DRA-DRP*DRDIA*.2-.666*TOOLW +SETFO-DRMWT
      URACC=URTHR/URMAS
      DRACC=DRTHR/DRMAS
      URVEL=URVEL+URACC*DT
      DRVEL=DRVEL+DRACC*DT
      URSTR=URVEL*DT+URSTR
      DRSTR=DRVEL*DT+DRSTR
      URDIS =URSTR*URA
      DRDIS =DRSTR*DRA
      URP=URFP*(VOL/(VOL+URDIS))**1.4
      DRP=DRFP*(VOL/(VOL+DRDIS))**1.4
      IF (URSTR+DRSTR-STR) 3,3,4
    4 RELVE =URVEL+DRVEL
      EN=((.5*URMAS*URVEL**2.)+(.5*DRMAS*DRVEL**2.))/12.
      WRITE(1,5)DT
    5 FORMAT(48HCBLOW SINGLE UPPER AND LOWER RAM VELOCITY STUDY,2X,
     110HTIME INTEG,F6.4,3HSEC//)
      WRITE(1,6)URDIA,DRDIA,STR,VOL
    6 FORMAT(5HURDIA,F4.0,1X,5HDRDIA,F4.0,1X,13HSTR TO IMPACT,1X,F5.1,
     11X,8HFIRE VOL,F8.0/)
      WRITE(1,7)URMWT,DRMWT,TOOLW,SP,SD
    7 FORMAT(5HURMWT,F7.0,1X,5HDRMWT,F7.0,1X,6HTOOLWT,F7.0,1X,6HSET PR,
     11X,F5.0,1X,7HSET DIA,F5.2//)
      WRITE(1,8)
    8 FORMAT(9X,4HFIRE,3X,5HFINAL,2X,6HSTROKE,1X,5HVELOC,1X,6HIMPACT,1X,
     13HNET,1X,6HENERGY,2X,4HTIME)
      WRITE(1,9)
    9 FORMAT(10X,2HPR,5X,2HPR,12X,2HIN,4X,4HTIME,14X,9HINCREMENT)
      WRITE(1,10)
   10 FORMAT(10X,3HPSI,3X,3HPSI,6X,2HIN,4X,3HSEC,3X,3HSEC,3X,6HFT LBS,
     16X,4HSECS/)
      WRITE(1,11)URFP,URP,URSTR,URVEL,T
   11 FORMAT(9HUPPER RAM,F6.0,1X,F6.0,1X,F6.2,1X,F5.0,1X,F5.3)
      WRITE(1,12)DRFP,DRP,DRSTR,DRVEL,T
   12 FORMAT(9HLOWER RAM,F6.0,1X,F6.0,1X,F6.2,1X,F5.0,1X,F5.3)
      WRITE(1,13)STR,RELVE,T,EN,DT
   13 FORMAT(6HTOTALS,17X,F6.2,1X,F5.0,1X,F5.3,2X,F9.0,3X,F6.4)
      CALL EXIT
      END
```

DISCUSSION OF RESULTS

Our first print-out gave velocities and energy approximately within the specifications for 1500 psi fire pressure.

It became evident that the assumed ram weights were slightly excessive in relation to our efforts to attain specification velocity. We now felt that we must study the shaping of the counterblow ram system with an eye to reducing unnecessary weight and so attain the highest velocity possible to enable us to comply with the specification.

At this time we had an indication that our fire volume was excessive and it remained for future Program III to develop further machine parameters.

We also developed a program concept for determining exact cushion system parameters, Program IV, by rebuilding the Fortran of Program I to take into account high fluid flow to the cushion systems and an exact accounting of cushion thrust with High Reynolds Number.

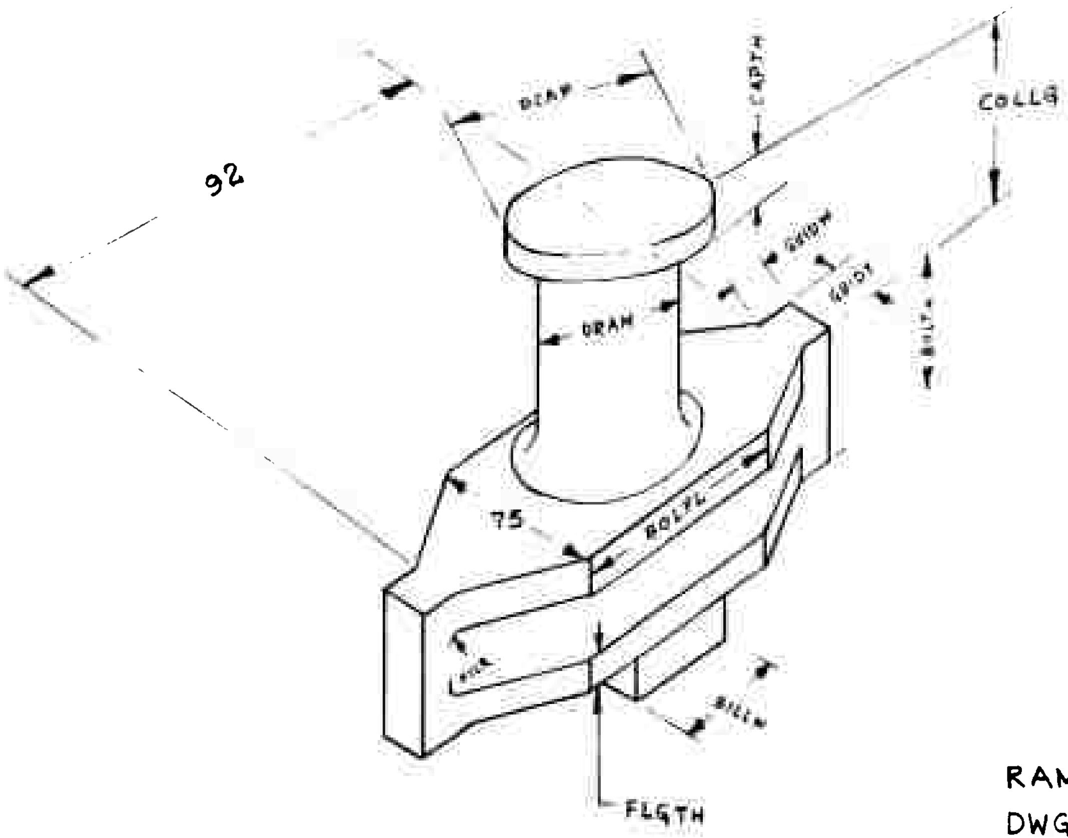
To fully understand the input dimensions, refer to drawings "Dimension Key" and "Load and Moment Arm Key," AA-10521 and AA-10522.

Although bolster section was set up as an I-Beam, it can be programmed as a solid bolster by merely making the flange thickness "FLGTH" equal to 1/2 the bolster thickness "BOLTH."

The program will compute the weight of any size column and bolster assembly and give breakdowns of the weight distributions.

The abbreviations are almost self-explanatory on the weight breakdowns.

FLGWT	-	Flange weight - bolster I-Beam
WEBWT	-	Web weight - Bolster I-Beam
GUIWT	-	Total weight of guide portion
FILLWT	-	Weight of fillet radius blend portions
HOLE	-	Hole in column if used; if not used, input - 0
TOOWT	-	Tool weight (This effects inertia stress and velocity for any given Energy Input.)
BOMIX	-	Moment of inertia of the bolster section. (Uses the 75" ram width and computes the cube of the "BOLTH" depth of ram taking into account I-Beam design when applicable.)



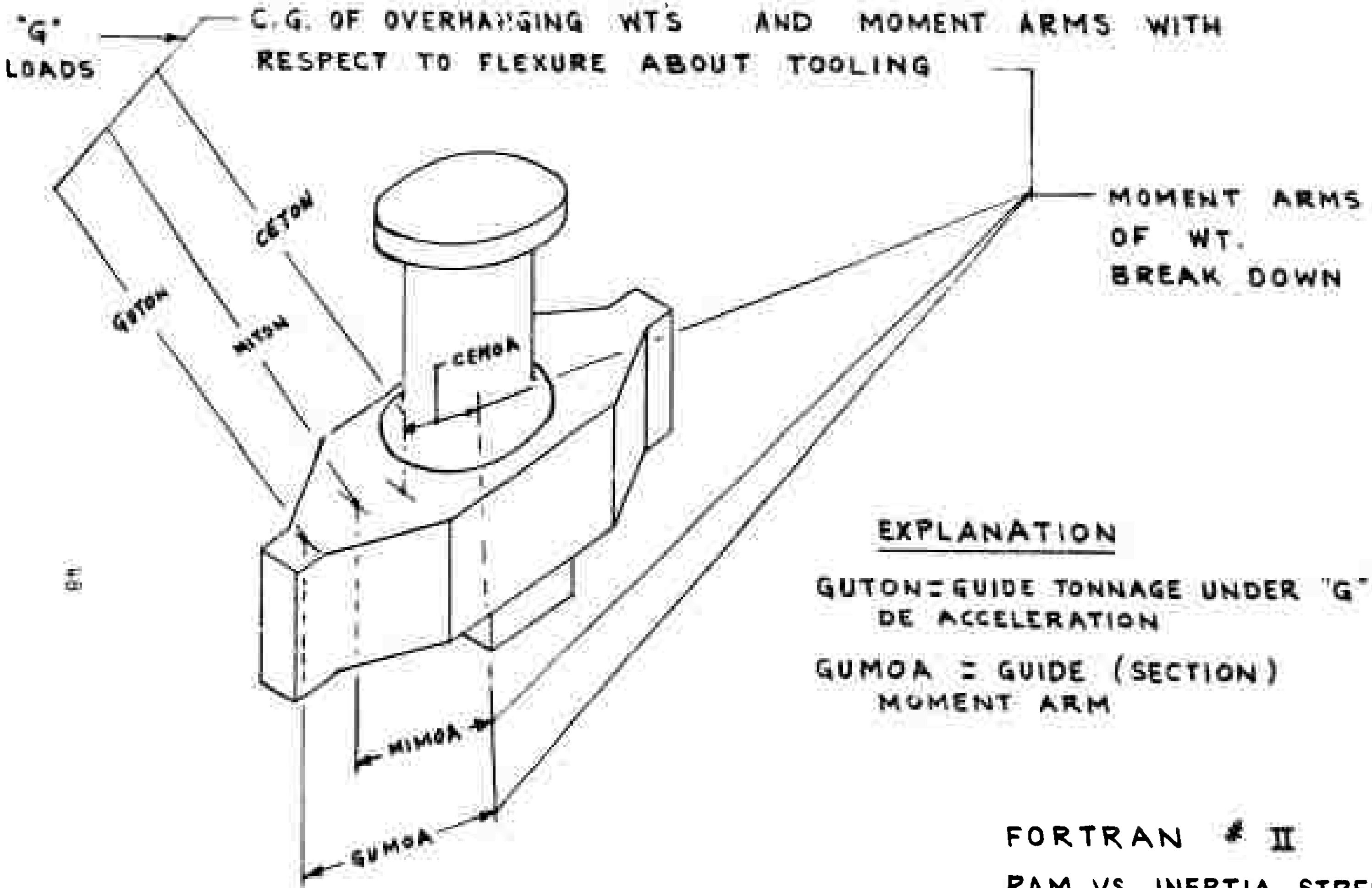
FORTRAN # II

RAM VS. INERTIA STRESS
DWG. DIM. KEY

DATE 4 - 15 - 69
BY: J. KOSKORIS

PLATE XV

DWG. No. AA - 10521



EXPLANATION

GUTON = GUIDE TONNAGE UNDER "G" DE ACCELERATION

GUMOA = GUIDE (SECTION) MOMENT ARM

FORTRAN # II

RAM VS. INERTIA STRESS LOAD & MOM. ARM KEY

DATE 4-15-69
DWG. BY J. KOSKORIS

PLATE XVI

DWG. No. AA-10522

COMPUTER PROGRAM II

Fortran Plan Descriptive Notes on
Program Studying Ram Weight Vs. Inertia Stress

Ram Diagram Included Depicting Moment Arm
and Force Abbreviations Used in Fortran Plan as
Dimensions

General Discussion of Printout Data

Particular Discussion of Printout Data

General Discussion

Basic relationships between flexure stress, FLEXS, in the critical section of the rams were studied as they varied with severe hard shots. A "hard shot" being a blow with a small amount of metal deformation, and a high number of (G)s "DEACCEL." The smaller the deformation, "STOP," the harder the shot and the higher the flexure stress of the portion of the rams beyond the workpiece, "BILLW."

This flexure stress, which is the limiting criteria of any high velocity machine, determines a safe upper level of operation with respect to the hardness of the shot or severity of the blow. This is usually discussed in terms of "G" level which is the primary reaction of the rams to the forging load and is printed in the first column of output sheet.

This program for any particular set of dimensional inputs and die dimensions yields an output of varying tonnages. These tonnages or tool reactions are then related to "DEACCEL (G)s" - Deceleration 'G' level, "FLEXS" - Flexure stress which is limiting, "CSTRE" - Compressive stress due to the weight of the column, "DEACC" - Deceleration time, and "STOP" - actual metalforming stroke required to generate the particular load at any particular impact velocity printed above "STOP."

Particular Discussion

Note that in Printout II_2, negative moments give rise to an unnaturally reduced flexure stress. Data inputs were corrected to yield more conservative values of flexure on Printout II_3.

Printout II_3 is a realistic run of a solid ram operating at a reasonable impact velocity. Note that velocity has been reduced from 350 to 320 inches per second per each ram.

Note that field of the maximum angle has been increased to three places to show elastic curvature of the bolster section under high 'G' loading. The small deflections shown are due to the weight-moment relationship of the portion of the bolster extending beyond the assumed width of tooling of 45". The previous run, II_2, was based on wider tooling width of "BILLW" - 50". This was the principal data change that resulted in elimination of the negative moments on Printout II_2 previously discussed.

Summarizing, Program II yielded significant results. Using the outputs of Program I, ram diameters and impact velocity, and varying ram bolster thicknesses and plan view dimensions as inputs, we investigated key stress and deflection resulting from soft to extremely hard blows. The range investigated was from 250 to 2,250 G's, corresponding to tooling tonnages of 9,424 to 84,818 at impact velocity of 640 to 700 inches per second. Note that at 1,000 G's., the stress levels in flexure of the solid ram, exhibit 2, is a safe 10,387 psi with an assumed billet or die width of 50". The corresponding tonnage that is produced is 37,697 with a total metal deformation of twice .158" or .316". It is interesting that this blow takes place in the 5/16" of metal movement in approximately 1/1,000 second at an impact velocity of 700 inches per second.

What is particularly attractive about this program is that any bolster configuration can be readily programmed so that resultant weights of bolster and column are machine calculated along with the resultant stress's produced on the rams when hitting a particular forging of known width, "BILLW."

The weight moment effects of the bolster were subdivided and printed out so that an unusually bad condition of overhang could be pinpointed quickly. The overhang of the ram was divided into three parts. The central section having parallel sides was considered with respect to the widest portion of tooling designated

"BILLW." Then if there was a positive moment of this section causing flexure it was computed and its Moment Arm about "BILLW" was designated "CEMOA." A negative sign indicating that the width of the parallel sides was smaller than "BILLW," e.g. BILLW - 50 and BOLFL - 47 1/2, a minor negative weight moment occurs which we first thought tends to reduce the flexure moment stress by a minor amount. Errors resulted due to improper Fortran interpretation of this negative moment. The errors resulted in an artificially reduced flexure stress readout labelled "FLEXS" on the printout worksheet which we felt was not conservative. Exhibits are included to show the reinterpretation of this negative moment and its effort on the printout flexure stress. Note that in exhibit F-3, the negative moment sign is deleted from the breakdown of weights and their moments and the resultant flexure stress level prints out at a higher value which we feel is more conservative interpretation.

That leaves the guide portion and the intermediate gradually tapering portion to be considered. The weight of the guide portion and its respective flexure moment arm about the extreme edge of the tooling are designated "WTGU" and "GUMOA" respectively. Counterpart quantities for the intermediate section are designated "WTINT" and "BIMOA".

The flexural resistance to both stress and deflection is proportional to the moment of inertia of the bolster at the critical edge of the tooling. This value along with a complete breakdown of all inputs and computed weight distribution for the ram column assembly are printed out by the computer in printouts for Program II denoted II_1, II_2 and II_3.

COMPUTER PROGRAM III

Plan and Review of Final Results

Machine Design Stress

Analysis of Key Members

Review of Overall Results Along with Explanations of "Soft" Items Not Directly Answered by Computer III_3 Printout

Detail No. I Column

After many trials a column ram and cap diameter of 24" and 36" was chosen. The ram diameter was principally chosen as a result of kinetics studies in I and IV to accomplish rated specification parameters of energy and velocity with pressures conservative enough to assure results. If required, pressure levels could be increased to 2000 PSI which has been past Dynapak criteria for fire chambers.

With a total weight of ram and bolster of 70,000 pounds (determined as reasonable from programs I, II and IV), the critical pressure in the upper fire chamber needed to hold the upper ram in balance is 123 PSI. The complex stress in this cap corresponding to an approximate 35,000 ton blow were investigated and found to be a reasonable approximate 17,000 PSI stress due to flexure. Many cap thicknesses were evaluated before choosing one for the best combination of strength to weight as evidenced by 1000 'G' flexure stress. Studies indicating the potential column weight saving with hollow construction indicate 55% reduction possible. Such weight savings cannot be ignored when studying the compressive stress at the root of the column at rated 1000 'G'. Note that with the hollow construction the compressive 1000 'G' stress at the base of the column taking into account the cap is a reasonable approximate 17,000 PSI.

The first column connections studied were two member column and separate bolster designs. An early program in this III group was run for a column clamp plate secured with (20) 4" studs. Both clamp plates, detail number 4, reached an excessive weight of approximately 11,000 pounds based on a 9" thickness. This design takes into account leverage edge effects and assumed full 1000 'G' reversal based on capability to take particular hard shots. Joint was designed to transfer eccentric loading up to 10,000 PSI flexure in the major column diameter and the excessive weight arose mainly for this consideration.

A program for a (4) tooth bayonet lock was considered and preliminary Fortran Program written but was discarded in favor of a sow key lock. The sow key lock being similar in design to conventional hammer practice used to lock tooling to beds.

The sow key design was successfully programmed and proportions of an acceptable design were determined. Design criteria here was that key should withstand a full 1000 'G' reversal. This criteria is conservative since strain gage testing of this portion of the one piece column on our 1220 and 620 have never shown any major 'G' reversals. Reversals only occur at sections subject to flexural beam 'G' stresses at critical junctions of Dynapak frames. A 1220 C test shot involving over 100,000 foot pounds with less than 1/64" deformation was strain gaged and showed only a simple compressive stress at the junction.

During the computer program, a major forging vendor furnished us with a quotation for a one piece ram-column. By having a better alternative, we now abandoned further computer studies with the sow key ram lock in favor of a simple one piece forging.

<u>X-Beam Detail No. III</u>

The upper and lower X-beams are fabricated from rolled 18" thick mild steel slabs and machined to 17" thickness. Note under maximum thrust corresponding to a 1.3/1.0 overload ratio maximum flexure stress developed in these X-beams equals approximately 10,500 PSI. This calculation takes into account the anchoring pattern of the main side frame studs and the centroid of the fire chamber reaction tierods which transmits the thrust reaction to the X-members.

The simple tensile and hoop stresses in many of the machine members were calculated based onl.3/1.0 over-pressure load. Stresses were based on actual root area and simple hoop stress criteria where $S_t = \frac{\text{I.D. x Press.}}{\text{2 x Wall Thick}}$

Detail Stud No. VII

With regard to the studs anchoring tie plates to side members it was felt that many small heat treated studs would be better than a few very large tierods and nuts running through the entire length of the structure. Poor experience on early Dynapaks with large nuts that became frozen to tierods after a year of service affected this decision.

The initial series of Fortran computations indicated that our first sizing choice was too small however. Hand computations of the stud loading due to leverage after increasing the number of bolts in the extreme leverage now reduced the flexure stress in the 3" - 8N alloy stud from 90,000 PSI in the smaller fastener to 17,000 PSI at rated 1500 PSI fire pressure after increasing both the size and number of fasteners in the extreme row. With this change, we utilize (18) 3" studs per side member so that the direct tensile stress due to maximum rated thrust is only as follows: (Reference drawing 10523/A).

At 2000 PSI F.P. thrust on each side member equals 452,159 pounds with a 24" diameter ram.

Root area of 3" - 8N stud equals 6.6 inches based on 2-85" root diameter.

$$S_t = \frac{452{,}159\#}{(18)(6.6)} = 3{,}800 \text{ PSI}$$

It is clear that the simple direct stresses even at overload of 1.3/1.0 are insignificant considering the strength of the alloy - heat treated studs and helicoil inserts used to tie down the basic frame components. To get a high efficiency joint between the end members and side slabs (6) 4" diameter dowels were used on each slab connection (Reference 10523/B). Since these dowels have generous bearing surfaces and are made from high hardness shock resistant steel assume an allowable shear stress of 20,500 PSI to develop resistance to side loading. The total side resistance of the dowels not counting frictional drag of the main fastener end studs would be equal to -

$$S_{side} = (12)(10\ 1/4 \text{ TSI})(4^2 \times .784)$$

$$S_{side} = 1{,}550 \text{ Tons}$$

The total frictional load due to a 20,000 PSI prestress on the (36) 3" studs clamping the end plates to the side member assuming a friction factor of .3 is -

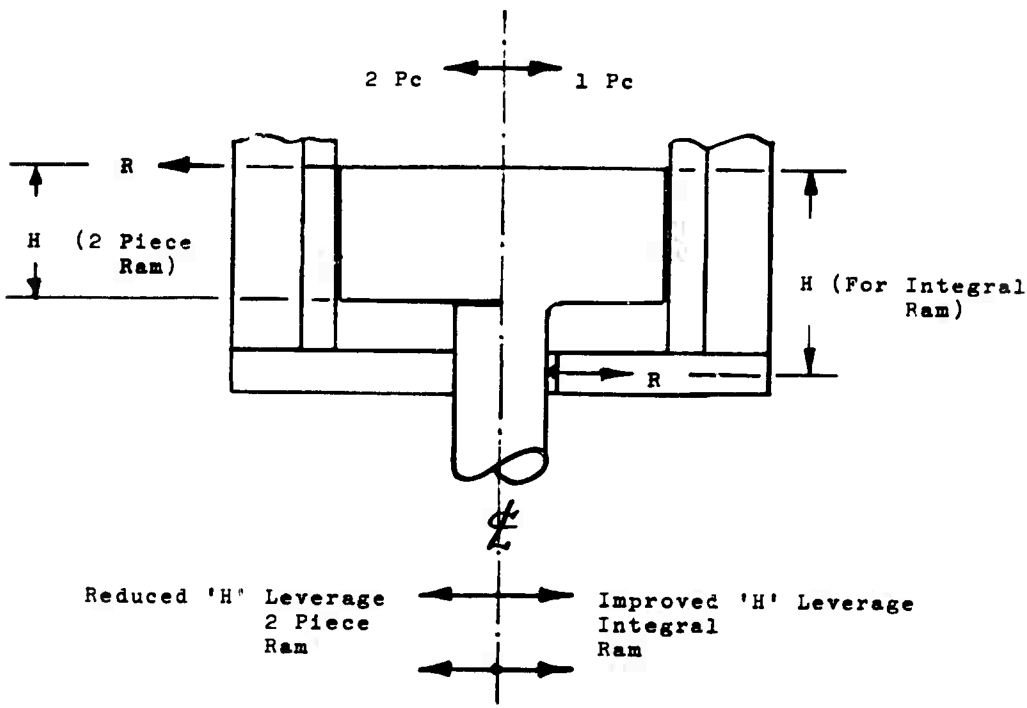
$$F_{r\ side} = (36)\ 6.21 \times 10\ \text{Ton/in.}^2 \times .3 = 680\ \text{Tons}$$

The total resistance to side loading between the end plates and side members is approximately 2,230 tons taking into account reasonable factors of safety.

To be conservative, neglect the frictional side load capability of the main anchor studs above. Consider what a dowel side resistance of approximately 1,500 tons means in terms of eccentric forging loads. Note that half this load is put into each - solid slab tower.

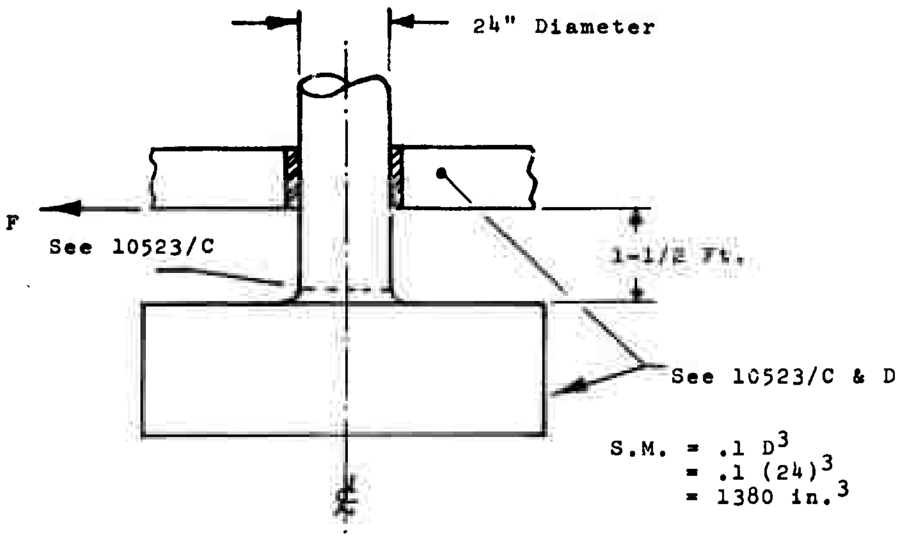
Bending Considerations and Side Load Capability

Because integral rams-columns are utilized they tend to reduce point loading or the gibs for off center loading.



In any c'blow machine reaction moments of upper and lower ram systems will be equal and opposite and the fixed structure is called upon to react these moments within the gib guiding arrangement. Excessive gib bearing stresses will cause stick slip under load and so impair the efficiency of the process. It is for this reason that the integral ram design was chosen for both rams. The tremendous section modulus will reduce gib loading without increasing the weight of the rams. The only other method of reducing gib loading under eccentric conditions is to make the height of the box section much longer. Note that with the proportions in our design a 36" deep section even when contoured at the perimeter where it is unnecessary and undesirable caused a resultant closing velocity lower than contract specification with the specification proportions of plan area.

Working backwards assume a one foot stroke of each ram prior to impact. Now stress the column in flexure to a safe maximum value, say one half the minimum yield strength. For a maximum working stress of 40,000 PSI what moment can column sustain.



$$\text{S.M.} = .1\ D^3$$
$$= .1\ (24)^3$$
$$= 1380\ \text{in.}^3$$

A 24" diameter ram has a section modulus of 1380 in.^{3} The maximum force 'F' it can sustain is based on simple flexure.

$$40{,}000 \text{ PSI} = S_b = \frac{M}{S.M.} = \frac{F \times 18"}{S.M.}$$

$$\text{Solving for } F = \frac{1{,}380 \times 40{,}000}{18} = 3{,}050{,}000 \text{ Lbs. or } 1{,}525 \text{ Tons}$$

Now consider the effect of this tonnage acting as a couple to the furthest extreme gib shoes. The bearing pressure of these shoes under this arbitrary side moment load will be as follows on the 11 x 5 1/2 area sets of gib shoes.

$$S_b = \frac{762 \text{ Tons}}{5\ 1/2 \times 11} = 12.9 \text{ TSI}$$

The bearing area in the main bushing under the same loading will be -

$$S_b = \frac{1525}{18 \times 24} = 3.6 \text{ TSI}$$

Making the shoes much longer would not help due to the elastic conditions in slabs under the action of these huge loads.

To insure low friction and minimum wear under relatively high gib pressures hard facing materials of differing hardness and alloy are to be plasma sprayed on both the gib rail and ram wear shoes. These surfaces will be 56-61 Rc Colmonoy #6 chrome boron nickel for the rail and 45-50 Rc Colmonoy #5 somewhat lower alloyed than #6 for the shoes. Their inherent controlled porosity will allow the retention of high pressure lubricants such as $MO\ S_2$.

Getting back to the stresses under the maximum flexure condition. Next consider the means of transmitting lateral side load from the 18" thick end plates to the 26" thick side posts. Although we utilize (36) 3" studs in each end plate to make the connection to the side posts, neglect the considerable side friction

developed by these studs. This friction will be approximately -

$$F_r = (36)\ 6.2\ \text{in.}^2 \times 10\ \text{Ton/in.}^2 \times .3\ \text{in.} = 680\ \text{Tons}$$

The side load neglecting this friction will produce a shear stress of 10.2 TSI on the (12) 4" diameter solid alloy dowels installed to get great rigidity at the principal corners of the structure.

$$S_s \ \text{Dowels} = \frac{1525\ \text{Tons}}{(12)\ 4^2 \times .785} = 10.2\ \text{TSI}$$

To complete the stress picture in the 26" thick side slab consider the effect of this load. Each nominal side slab has a section modulus of 9,200 in.^{3}

$$Z = \frac{bd^2}{6} \qquad b = 82 \quad d = 26$$

The resultant flexure stress on the side members approximately 4' down from the top is 8,000 PSI.

$$S_b = \frac{M}{Z} = \frac{762 \times 2000 \times 48}{9{,}200} = 8{,}000\ \text{PSI}$$

Choice of 762 tons, half the total side load of 1525 tons, was due to the action of both vertical slabs to divide any side load transmitted through the top ram bushing by the 24" diameter column.

Note that for eccentric loading 90° away from the axis discussed, the side members will be stiffer and stronger by the ratio of - $\frac{28{,}600}{9{,}200} = 1{,}3.15{:}1$. Therefore, for the example previously cited the flexure stress in the simple side post will be $\frac{8{,}000}{3\text{-}15}$, only 2,500 PSI in the strong axis.

III Plan
Explanation of Fortran Plan for Machine Design

The latest Fortran Plan is included here for reference. The scope of this machine design task was very wide. Note that this Fortran is written for a two piece ram-column with a sow type of key.

Prior to this design, the present Fortran was written for bolted gland type of connection which proved too ponderous to be practical.

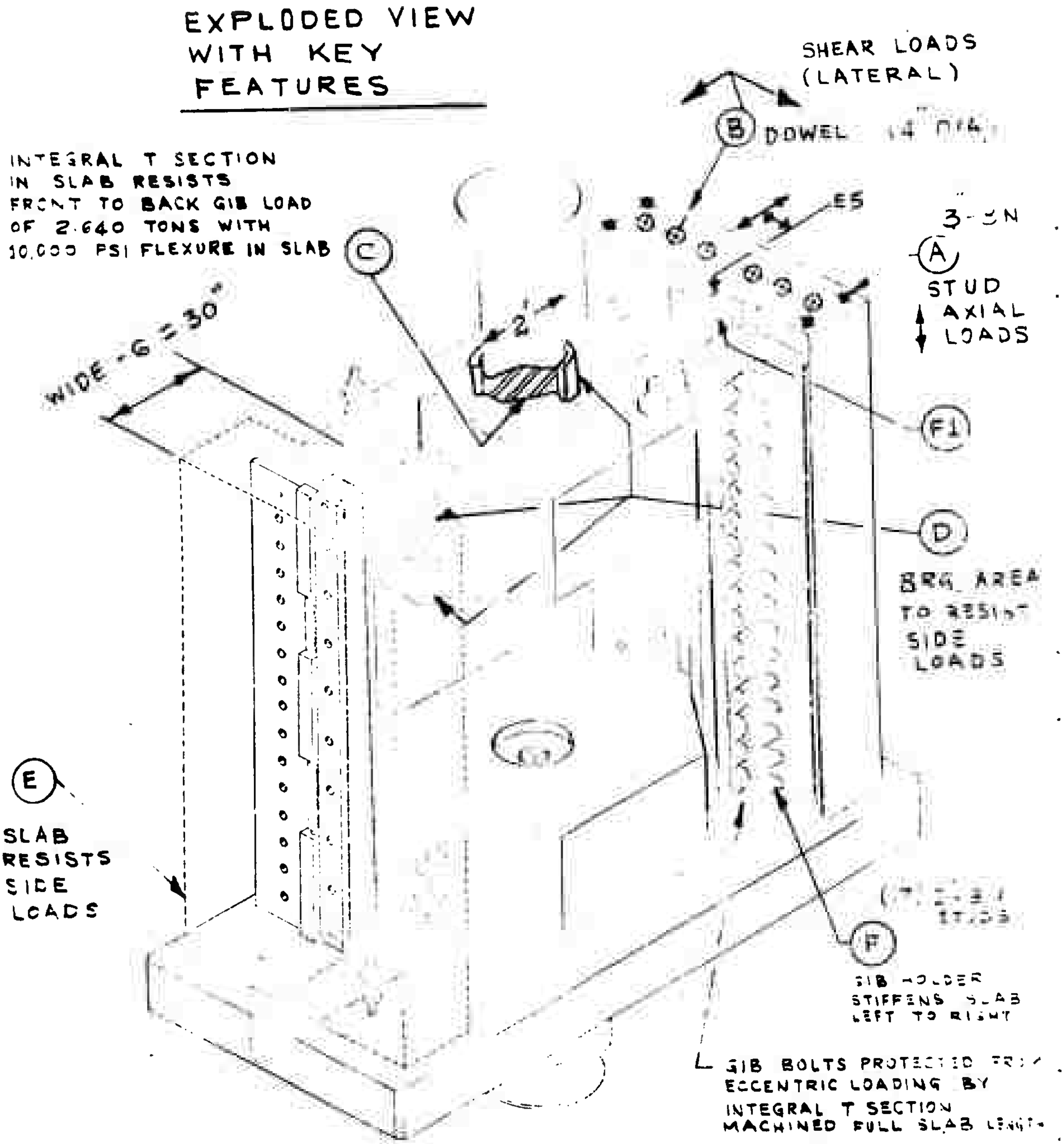
Even the sow key lock which we have fully programmed and proportioned is far inferior to an integral ram design. When it was determined that a one piece integral ram-column was practically feasible, we abandoned all thought of any other alternative. No further Fortran was programmed and hand computations of design relationships of the one piece ram-column to the total machine environment were then made.

Summarizing

Fortran expressions for Program III attempted to store as many of the designers key input specifications as possible and relate them to critical stresses imposed by a combination of static pressures and dynamic inertia forces. With this scheme, it was felt that danger areas could be pinpointed early in the program.

List of Key Design Statistics

(Refer to Drawing 10523)



GENERAL DYNAMICS
Electro Dynamic Division

PLATE XVII

DWG. No. 10523

DYNAPAK

1½ MILLION FT. LB.

FORGING MACHINE

MODEL 2436

		COMPUTATION	VALUE
A	Net clamping area of (36) 3" studs (root diameter undercut) to 2 13/16 diameter.	(36) 6.2 in.^{2}	225 in.^{2}
B	Shear area of main structure dowels (12) 4" diameter in both principal planes.	(12) 12.566	150 in.^{2}
C	Section modulus of ram at base (before radius) 24" diameter	24^3 (.1)	1380 in.^{3}
D	Total side bridge area (4) gibs and (1) ram bush. (Same in both principal planes)	(4) x 5 1/2 x 11 = 240 17 x 24 = 410	650 in.^{2}
E	1. Section modulus of side post (weak axis) bd^2	b = 82 d = 26	9,200 in.^{3}
	2. Section modulus of side post (strong axis)	b = 82 d = 26	28,600 in.^{3}
	3. Vertical load area	26 x 82	2,150 in.^{2}
	4. Axial stress in (3) due to 2000 PSI thrust on 24" ram.	$\frac{452 \times 2000}{(2)\ 2{,}150}$	210 PSI - Note unusual stiffness and stability as a guide housing.
	5. Shear area of T-gib retainer	3 x 170	510 in.^{2}

		COMPUTATION	VALUE
F	1. Shear area of T-gib retainer.	2 1/2 x 156	390 in.^{2}
	2. (17) 2" - 8N gib bolts area at root diameter	1 13/6 root area = 3.58 in.^{2}	44 in.^{2}
	3. Side post key (bearing area)	1 13/16 x 36" x 3/side (takes into account chambers)	200 in.^{2} single side post

The sturdy axis of the machine is actually front to back rather than side to side, due to the 3 fold stronger and stiffer side posts in this direction. The dowels 'A' off course are equally strong in all planes and so transmit their loading equally well front to back.

Consideration of eccentric loading front to back leads to an examination of how the gib holders (F) are retained by a T-head integrally machined into slab E_5. Shear area E_5 in the side posts has been purposely made stronger than its mating gib ledge F_1, 510 in.^{2} shear area side post versus 390 in.^{2} for the gib. The proportions of these tounges have been selected to minimize flexure stresses and allow relatively higher criteria determine design load capability.

Allow for the sake of visualizing what is going on, stressing of the (17) 2" gib retainer bolts to a conservative tensile stress of 10,000 PSI at their root diameters.

Therefore, on their pitch line one row of gib bolts will safely resist a force of 10,000 PSI x (44 in.^{2}) 440,000 pounds. Upon studying chart I, it will be noticed however that the center of the gib taking the loading from the ram has a greater moment arm about the corner, 19 1/4", than the bolt leverage of 10 3/4". Therefore, the bolt moment is reduced to, $\frac{10\ 3/4}{19\ 1/4}$ x 440,000 pounds, 245,000 pounds at the gib load line. Summarizing this sustains 245 tons by use of the gib bolts on two side posts against front to back loading, which is the strong side of the side posts.

Now consider the built in T-ledge which have a favorable moment arm relationship towards loading in the front to back direction. From chart II F-1 permit 6000 PSI shear to be applied to an area of 390 in.^{2} per side.post. Ton reaction is then, $\frac{6000}{2000}$ x 390, 1,170, which taking into account the favorable arms causing turning of the gib corresponds to, $\frac{33\ 1/4}{19\ 1/4}$ x 1,170 tons/gib x 2 gibs, 4,000 tons. Adding the slight helpful effect of the gib bolts gives a safe resistance to front to back movement of 4,245 tons.

This brings us to a consideration of how this load will be transmitted into the side posts; Chart II yields the bridge area of this side post key F-3 and from this area we compute the maximum frame bearing stress under the 4,245 front to back load as, 4245 x $\frac{2000}{(2)\ 200}$, = 21,230 PSI. This is a very reasonable bearing stress and a thicker key with a deeper groove in the side posts and gib retainer would merely increase flexure stresses in the notch without any useful result. Longer keys of course are possible but the present arrangement with the (3) short lengths of key facilitate key and gib removal and access to replace the main rams.

Summarizing Design Details from Previous Text and Referencing Them to Artwork

(A) 18/conn. 3"-8N alloy steel studs 4340 35-38 Rc total separating resistance area based on (36) x 2 13/16 root areas equals 6.21 in.^{2} x 36 = 225 in.^{2} with relatively small alloy studs having minimum tensile strength of 200,000 PSI it would be safe to stress studs to 50,000 PSI at which stress they would have a capability of resisting, 225 x 25 TSI, 5,700 Tons. The maximum thrust of our actuator at 2000 PSI fire pressure being only 452 tons. Therefore, it can be seen that the machine has a considerable margin of reserve strength to accomodate undetermined eccentric forging loads and still maintain stiffness required for good guidance of rams.

(B) The (6) 4" diameter dowel per column offer equals resistance strength to unknown loading tending to cause translation and resultant loss of guidance system stiffness. Total shear area of (12) dowels is, 4^2 x .784 x 12, 150 in.^{2}, at a shear stress of 20,500 PSI these alloy dowels take a 1550 ton side load in any direction.

(C) The drawing depicts the large section modulus resisting rotating forces on the frame assembly. Note this section modulus is 1380 in.^{3}

(D) Each corner 5 1/2 x 11 area of 55 in.^{2} area x 4 sets side means that 220 in.^{2} bridge area for side loads on rails per ram.

(E) Weak axis section modulus of slab side frame equals 9,200 in.^{3}

(F) Note the manner of loading the slabs from the gib holders utilizes the full strength of all members by means of T-shaped projections of the slabs and rectangular keys axially located. The strength of the combination does not directly depend on gib fastener bolt (F) which merely affix this member to the slab sides (E). Slab T section impart a clamping moment to gib holders which is favorable leverage compared to gib loading of rams. Hence the ledge strength of slab T section will be increased by the ratio of their respective moment arms, 33 1/4/19 1/4, 1.7 to 1. The section modulus of the T shape (F1) is as follows -

$$Z = \frac{bd^2}{6}$$ b = length of gib holder = 156"
d = depth T section = 3

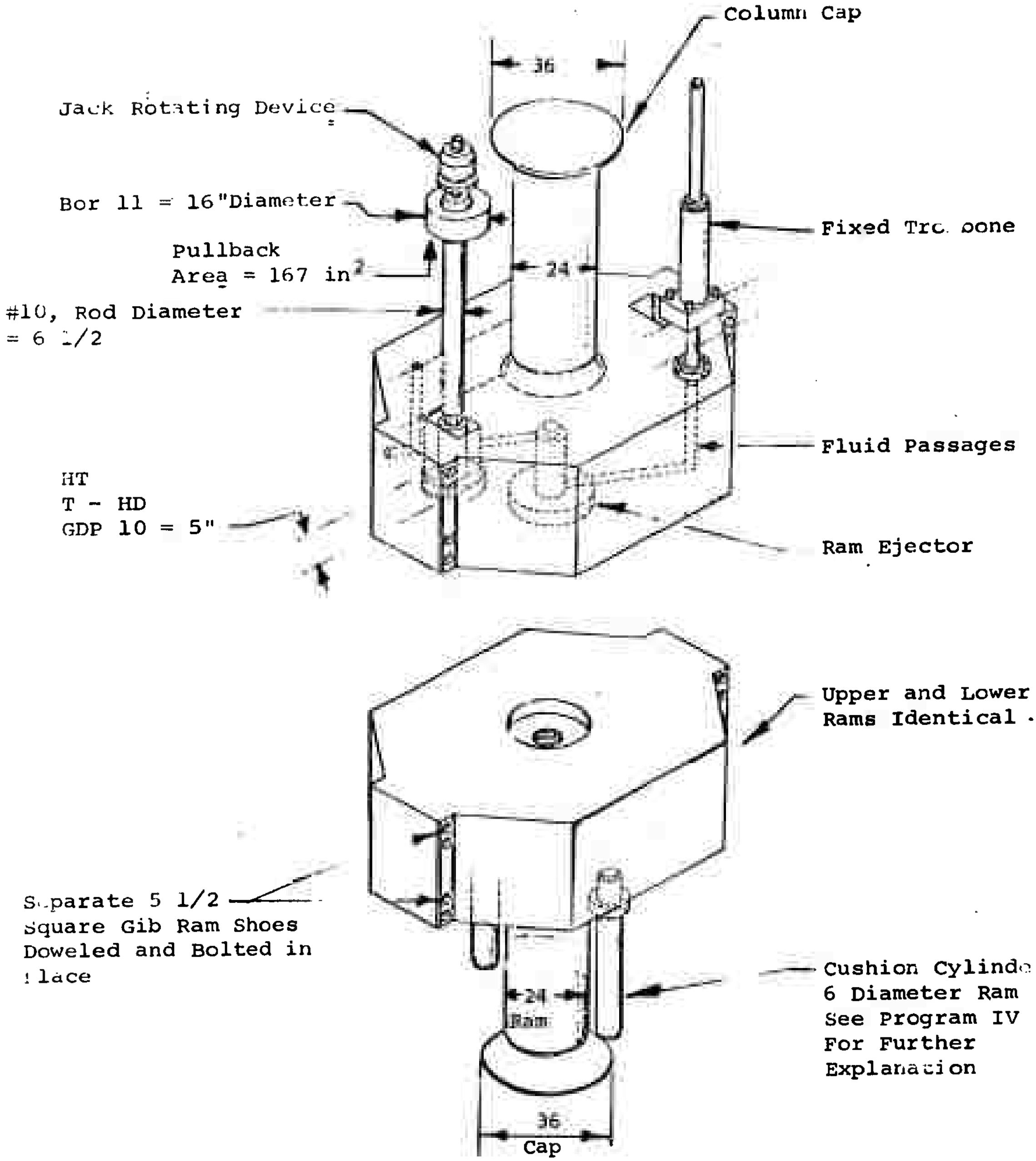
$$Z = \frac{156 \times 9}{6} = 235 \text{ in.}^3$$

The flexure moment arm is approximately 1/2 the ledge T projection equals 1 1/2". Therefore, at 10,000 PSI maximum flexure the ledge can withstand load P in tons.

$$10{,}000 \text{ PSI} = \frac{P \times 1\ 1/2" \times 2000}{235}$$

$$P = \frac{235}{1\ 1/2} \times 5 = 780 \text{ Tons}$$

Multiplying this ledge T-section safe load by the favorable leverage ratio we find that the equivalent load each slab can sustain is - 780 x 1.7 = 1,320 Tons. Since there are two side members the total front to back safe loading that can be applied to slabs (E) based on interlocking T-section is equal to 2,640 Tons. Of course this many alloy 2" studs attaching the gib holder to the slab will considerably add additional side load capability. However, since the integral T-section strength is so great, 2,640 tons with a factor of safety of 5, we prefer to consider that the T-section is used to prevent the gib bolts from seeing peak side loads can eliminate bolt maintenance attention due to stretching and loosening.

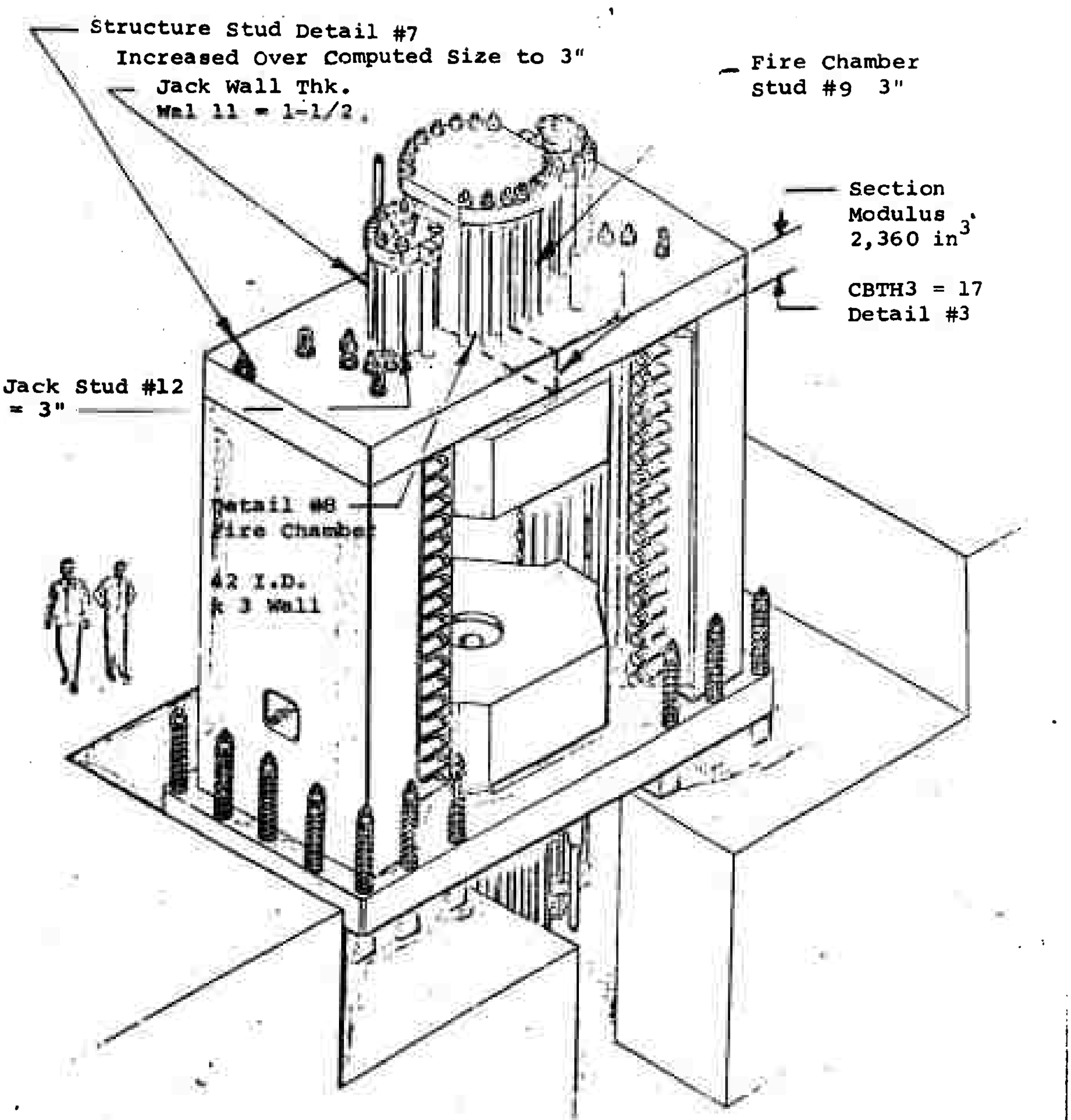


1½ MILLION FT. LB.

FORGING MACHINE

MODEL 2436

PICTORIAL READOUT
FINAL PROGRAM III_{3-2}

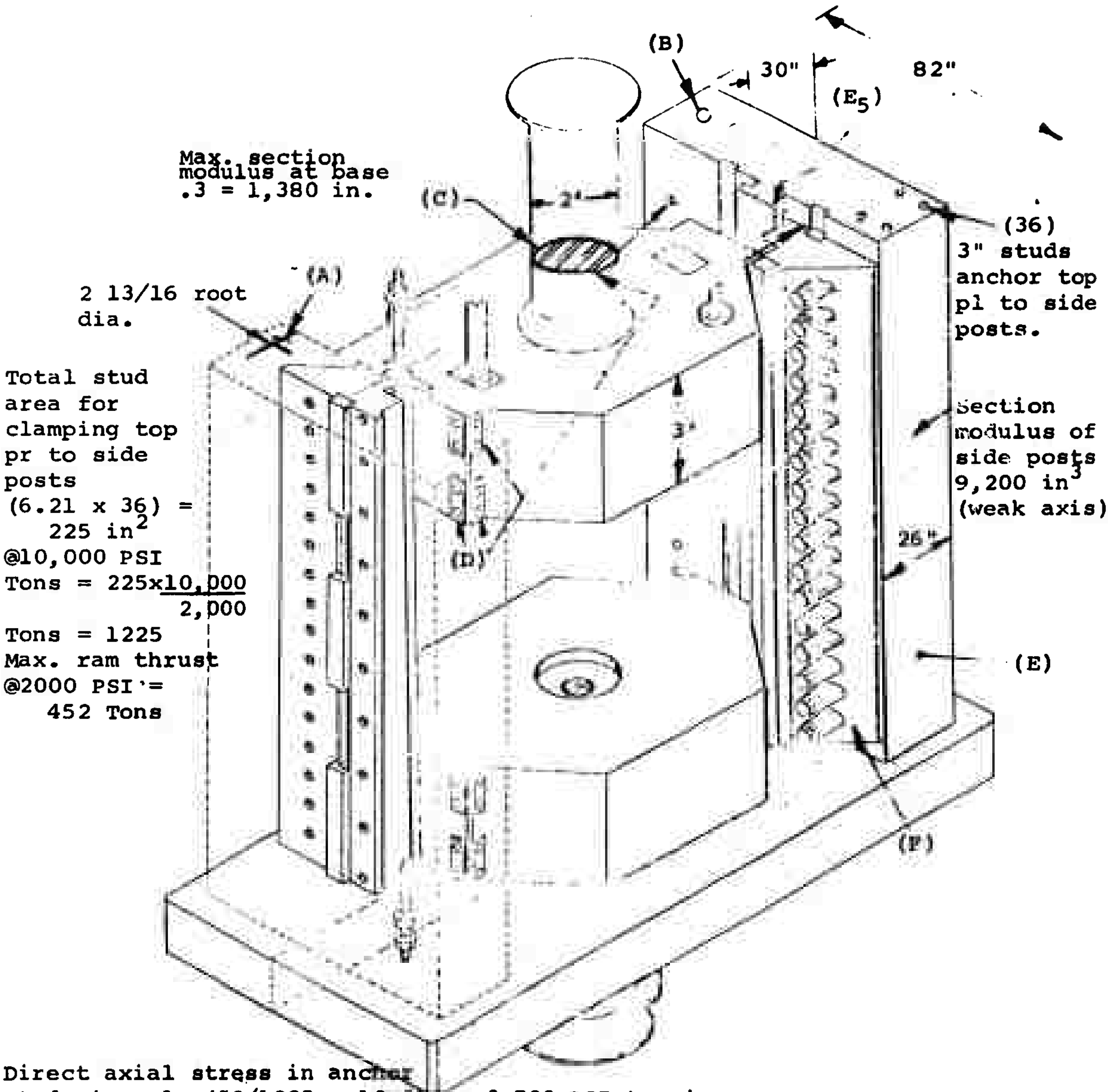


1½ MILLION FT. LB.

FORGING MACHINE

MODEL 2436

PICTORIAL READOUT
FINAL PROGRAM III_{3-3}



Direct axial stress in anchor studs is only 452/1225 x 10,000 = 3,700 PSI tension

1½ MILLION FT. LB.

FORGING MACHINE

MODEL 2436

CHART II

COMPUTER PROGRAM IV

Fortran Plan
Explanation
Dynamic Study

Note that the results are similar to Program I. Many of the Fortran cards from I have been used to form number IV program.

The I program has been modified considerably to show the more complex dynamics of the cushion system.

Inputs for the fire chamber system are identical to Program I. The input data, however, had at this time been brought up to date with the results of Programs II and III.

Several cushion cylinder diameter trials were selected before choice of 6" diameter cushion rams. Larger and smaller rams would not satisfy two important sets of criteria.

First the pressure drop (DPIR) between the cushion tank and rams must be a small fraction of cushion cylinder "set pressure." This is necessary to avoid cavitating the cushion system and so cause excessive air entrainment. And secondly, the cushion ram must have enough area to contain the after impact downward reaction with safe and reasonable pressures.

A more sophisticated program built on top of Program I was developed to better understand the physical parameters affecting the cushion system.

The following changes were made to Program I:

I - Program IV did not assume cushion pressure to be constant. Instead it computed pressure in the cushion cylinders as an output for any size cylinder and plumbing combination of I.D. and length, "DPIR" or pressure drop between cushion tank and cylinders was printed out for a simulated impact seen at intervals of 12/1000 seconds until impact.

II - Viscosity of the fluid, friction factor of the connecting pipe and Reynolds numbers during the very rapid advance of the fluid cushion system were taken into account in Fortran computations.

III - Friction on all U-cup seals was computed based on actual seal length, instant pressure and friction factor of .2.

IV - Conditions after impact were also investigated. Excess momentum and cushion pressures required to safely de-accelerate bolster were determined and printed out on worksheets.

The program also measured the downward movement of the rams immediately after impact. Pressures in the cushion system required to produce the arresting movement are input values to program and are printed out as "cushion relief setting." The time required to arrest after impact movement for any particular combination of cushion cylinder diameter and pressure was printed out on readout sheet.

The printout indicated that criteria are reasonably met. The maximum "DPIR" or pressure drop for the run was 338 PSI or approximately 56% of the initial 600 PSI in the cushion tank. On previous runs with an 8" diameter cushion cylinder "DPIR" was greater than 100% of cushion set pressure. For the 6" diameter rams, cushion pipe velocity was a reasonable maximum 71 feet per second and corresponded to a Reynolds number of 85,037.

Cushioning after impact which is the primary function of this system appears very attractive with the two 6" diameter rams.

A relief valve setting of 2500 PSI causes a deceleration of both rams after impact of approximately 1/3 of a 'G', resulting in a stopping distance of approximately 1" in 61 milliseconds of time.

Note that this program yields a maximum impact velocity of approximately 632 inches per second at 1,590,029 foot pounds approximate rated energy. The conservative aspect being that only 1300 PSI fire pressure was required to produce rated energy. The basic machine has been designed to safely accomodate 2000 PSI fire pressure in both the gas ram and oil hydraulic system.

Discussion of IV_2 Printout

This last dynamic run was made to bracket the capability of the machine. Note that fire pressure has been set for its maximum design limit of 2000 PSI. With cushion pressure set relatively high, 1000 PSI, to reduce gravity effects on the lower ram an energy output of 2,075,800 foot-pounds was developed taking into account a computer integrated allowance for U-cup seal friction during the adiabatic gas expansion in both fire chambers. Also note that both rams met very close to mid stroke so utilizing both jack systems efficiently. The machine still has additional reserve energy capability since there is the possibility of increasing the stroke an additional 5 1/2". However, it was felt that most forgings would have more ideal and faster cycles with a 30 1/2" total metal forming stroke. The percentage departure from ideal mid stroke impact being only $\frac{3/4"}{15\ 1/4"}$, 5%. It is interesting to compare the effect of cushion system relief valve setting on the arresting of both rams after impact. Note that a relief valve setting of 2,673 PSI (AIMTE) completely absorbs the kinetic, 6,391 foot-pounds and potential, 175,196 foot-pounds, energy after impact. Also note the unusually small excess kinetic energy that is available after impact, 6,391 foot-pounds, that is only capable of producing a downward after impact velocity of 20 inches per second to the two ram systems. When the relief in the cushion system is set at 3500 PSI the after impact downward motion is arrested after a travel of only 3/8" and takes only 38 milliseconds to come to a full stop. The maximum flow over the cushion relief valve immediately after impact is -

$$\frac{28.23\text{ in.}^2 \times 2\text{ cgls} \times 20''/\text{sec} \times 60}{231\text{ in.}^3/\text{gall}} = 292\text{ GPM}$$

To insure best performance two relief valves will be used, both of them having a 5000 PSI rating. A 1 1/2" direct acting Rexroth having unusually fast response will be the prime setting and it will be backed up by a 2" pilot operated denison, #RIV 32-535 set 50 PSI higher than the Rexroth in order to control the initial peak momentary surge. The supporting pipe system will be adequately designed for 5000 PSI to take momentary peaks that will rise above the basic setting of the cushion relief valves.

COMPUTER PROGRAM V

Hydraulic Accumulator System Computer Study

APPROACH

A computer study, Plate XIII, of hydraulic system characteristics, as Up and Down Jack time, to complete a 36 inch stroke was varied from 1 to 13 seconds.

ESTABLISHED DESIGN CRITERIA AS AN INPUT TO COMPUTER PROGRAM

From previous studies, the key dimensions of the oil operated jacks and gas rams were determined.

LIST KEY DIMENSIONS

1.	Gas ram diameter	-24"
2.	Gas ram pressure (to develop rated energy of 1-1/2 million foot-pounds)	-1400 psi
3.	Jack piston diameter	-16"
4.	Jack rod diameter	-6-1/2"
5.	Maximum forging ram stroke (Total)	-36"

COMPUTER PLAN (FORTRAN)

The plan was set up to take inputs listed above as key dimensions 1 through 5 and give an indication of hydraulic system characteristics for different rates of performance. The computer print-out would first give the characteristics for a 1 second time to make the complete stroke, and then add a second to the cycle and print out new outputs for the next cycle time, e.g. 2 seconds. When the stroke time reached 13 seconds, 13 steps of data, additional stored input data was printed out and the program terminated.

Additional input data that was printed out covered the following numbered inputs, 1 through 6.

ADDITIONAL INPUTS TO COMPUTER PROGRAM

1. Pipe friction factor — .025
2. Plumbing data — Pipe I.D. Pipe Length (Branch pipe Breakdown)
3. Oil cylinder piston and rod size
4. Gas ram diameter and pressure
5. Gas ram weight since the jack cylinder pressure is partially determined by it
6. Accumulator charge pressure

BASIC CONCEPTS USED IN THE FORTRAN PLAN

1. Pipe Pressure Drop - Based on empirical pipe data, friction factor and Reynolds number.
2. Cylinder Motion due to Accumulator - Based on NET pressure in cylinder after computing pressure drop above, balance pressure to overcome a gas cylinder load and F=MA.
3. "Lohm" Concept of Piping Resistance - Makes piping losses analogous to electrical resistance and so lends itself to simple analysis. Based on LOHM Laws of "Lee Co.," Westbrook, Connecticut.

PIPING NETWORK ALLOWANCE FLOW RESISTANCE FOR VARYING TIME

Permissible piping resistance in Lohms for the computed time and accumulator charge pressure. From the Lohm values printed out as ULOHM and RLOHM, a comparison of the actual piping losses in Lohms can be determined and judgment made about increasing or reducing the selected pipe size or increase accumulator charge pressure. Selecting pipe size is still a trial and error procedure but the computer print-out eliminates laborious hand computations involving network piping losses.

"TLOHM" OR TOTAL PIPING RESISTANCE PRINT-OUT MEANING IN RELATION TO "ULOHM"

Actual piping network losses based on network of input pipe diameters and lengths, friction factor and Reynolds Number was printed out as TLOHM. Since our computations assumed a constant value for friction factor, its value was constant in the print-out. NOTE that where the piping resistance determining retraction speed, ULOHM, exceeds the value of TLOHM, that time value is the limiting performance that can be obtained for the input conditions stated. In this case, a three-second time for Jack Return requires a piping resistance not to exceed .716 LOHM with a 3500 psi accumualtor charge and 1400 psi Fire Pressure in the gas cylinder. Since .716 LOHMS is only slightly greater than the .6 (TLOHM) value, the accumulator and plumbing system can effect jack retraction in three seconds.

(FORTRAN PLAN) REGENERATIVE VS. PREFILL CONCEPT

A study of the computer print-out sheet, fifth column, under the heading AGPMR indicates flow rates for accumulator rapid advance of the jack cylinder in a regenerating mode. This initial idea has since been discarded in favor of a simple low pressure prefill system to advance the jacks into the rams after firing for the following reasons.

DISADVANTAGES OF HIGH PRESSURE REGENERATIVE SYSTEM

1. Required additional high pressure valves and piping.

2. Consumed 10 gallons of high pressure accumulator fluid, thus reducing capacity for the main task of retracting the jacks against a gap resistance. The main task requires 52 gallons.

ADVANTAGES OF A PREFILL SYSTEM

By utilizing a low pressure nitrogen-fluid vessel to advance the jacks without a load, the above two objections are overcome and a much simpler circuit results.

FINAL CIRCUIT

The final circuit is similar to Plate VII but several modifications have been made as follows. The new circuit is shown on Plate VIII.

1. A common set of accumulators to control upper and lower jack systems eliminate considerable redundancy and assures reliability.

2. A single prefill vessel replaces the original concept using four prefill vessels instead of depending on the original short run for low pressure drop, giant pipes, 8-inch SCH .80 carry exhaust and prefill fluid back to prefill and gravity vessels located in a pump room below the floor. In comparing Plates VII and VIII, note that one prefill vessel replaces four and one gravity tank replaces two in the new arrangement.

WATER HAMMER OR FLUID INERTIA EFFECT

The best feature of the circuit on Plate VII has been retained in regard to having the main control valves 1, 2, 3, and their counterparts in the lower cylinder system 1A, 2A and 3A remain close to cylinder ports. By keeping these valves as close to their respective cylinder ports as possible, the inertia effect of rapid opening is reduced to minimum pressure override.

APPENDIX IV

Electrical Control System

I. **MAIN STEPPERS**

The automatic events are categorized as follows:

1. Main automatic sequence.

2. Jack rotary locks.

3. Prefill or jack rapid advance cycle.

4. Jack retraction cycle.

Each of these categories is controlled by a separate stepping relay, insuring that all the events in each category take place in the proper sequence. (See Charts 1 through 4.)

The following explanation covers each separate stepper.

1. IMT - Automatic

Note the automatic stepper coil and ready light on lines 20, 21 and 24. The ready light on line 24 indicates that all principal safety features and initial conditions have been met. The main safeties are that both rams are retracted, RLS-1, and the trigger vent pressure is open to atmosphere, RPS-TV, indicating a safe seal condition prior to rotating the jacks to clear the rams. Secondary functions regarding the main control pilot operated check are also monitored. If a pilot valve should mechanically stick, causing one of these valves to open out of sequence, the permission light would not advance to its No. 2 position on command, thus avoiding a possible malfunction.

From a study of each stepper logic cam chart, it can be seen that a particular function must take place before the stepper moves to its next number. For simplified trouble-shooting, all four (4) steppers will be located behind a transparent panel in view of the operator. In this way, a cycle arrest can be quickly pinpointed to its cause by noting the stepper numbers.

2. Locks, Prefill and Jack Retraction

The Locks (Chart 2), Prefill (Chart 3), and Jack Retraction (Chart 4) steppers sequence through their cycles based on proper events occurring at proper intervals. If a malfunction occurs, the stepper will remain in its last numbered position. The operator can then view the last numbered position through the transparent panel and take proper action for subsequent troubleshooting.

II. CYCLE COUNT CIRCUIT RELAYS R101 - R104

The heart of the electrics is the capability of multi-hit utilization. The operator can set his selection control for 1, 2 or 3 hits. The machine will then recycle and the respective ejectors will become effective only on the last hit.

In addition, the same count circuits will cut-in the main accumulators for an energy burst as selected from the control desk. Here again the operator can set the time periods of accumulator energy release by selector control from his desk.

III. EJECTOR TIMER CONTROL IN CONJUNCTION WITH MULTI-HITS

Selector Off-On on line 12 puts the ejector out of cycle if not required. If it is required, the count of number of hits selector must be set as desired. For example, for three (3) hits, the count circuit is so interlocked that a feed to the main Eagle timers, 192 TR's, line 175 and line 177, will only be energized if two qualifications are satisfied. The first qualification is that 4M-2 on line 12 is closed. This will only occur when the ram starts up.

However, one qualification is not sufficient to fire the Eagle timers. Therefore, the press will hit three times in succession before picking up the count relay R103 on line 174.

As the timer 1 TR times-out, an adjustable dwell occurs prior to ejection and subsequent to the jacks starting up. At the conclusion of this dwell, delay contact 4-5 on line 176 fires the second timer, 2 TR, in the same manner.

BULLETIN 910C-1
CIRCUIT DESIGN CHART
FOR BULLETIN 910 STEP SWITCH

1MT

AUTOMATIC SEQUENCE

SYSTEM OPERATION	CONTROL DEVICE TO ADVANCE TO NEXT STEP	STEP	LOAD SWITCHES ON STEP SWITCH CLOSED: X OPEN: O																			
			X	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
STAND BY	PB	1	X						X													
"READY"	INTERLOCKS	2	X	X					X	X												
LOCKS OPEN	2MT-3	3	X		X				X	X												
MACH. FIRED	RPS-TV&RLS-1 n/c	4	X			X				X												
PREFILL COMPLETE	3MT-4	5	X				X			X												
LOCKS CLOSE	2MT-4	6	X					X		X												
JACKS RETRACT	4MT-7	7	X							X												
VENT SEAL (TR VENT)	1MT TAP SW.	8	X							X												
HOME TO POS. 1	"	9	X							X												
"	"	10	X							X			X									
"	"	11	X							X		X										
"	"	12	X							X	X											
		13	X																			
		14	X																			
		15	X																			
		16	X																			
		LOADS		2 MT STEP SW.	R5 RELAY	3 MT	2 MT	4 MT	R4 RELAY	1 CYCLE - R101	2 CYCLES - R102	3 CYCLES - R103	RUN-OUT - R104									

TYPE NO. MT ____________

SERIAL NO. ____________

VOLTS 120 FREQ. 60

NO. CKTS. ____________ NO. OF STEPS ____________

STEPPING CONTACT SYMBOL "02" . . . ____________

ROTARY SWITCH SYMBOL "05" — 12 POINT ____________

16 POINT ____________

ENCLOSURE NEMA 1 . . . . ____________

BULLETIN 910C-1
CIRCUIT DESIGN CHART
FOR BULLETIN 910 STEP SWITCH

2MT

JACK ROTARY LOCKS

SYSTEM OPERATION	CONTROL DEVICE TO ADVANCE TO NEXT STEP	STEP	X	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
			LOAD SWITCHES ON STEP SWITCH — CLOSED: X OPEN: O																			
STAND BY	SM-L	1	X																			
"READY"	INTERLOCKS	2	X																			
OPEN LOCKS	RPS-LDO	3	X	X	X																	
" "	RPS-LSO	4	X	X																		
" "	RLS-3	5	X	X																		
	2MT TAP SW.	6	X	X		X																
STAND BY	SM-L	7	X	X																		
CLOSE LOCKS	RPS-LDC	8	X																			
" "	RPS-LSC	9	X																			
" "	RLS-2	10	X				X															
HOME TO POS. 1	2MT TAP SW.	11	X																			
"	"	12	X																			
		13																				
		14																				
		15																				
		16																				
		LOADS		R1 RELAY	STD TIMER	1 MT STEP SW.	1 MT STEP SW.															

TYPE NO. MT

SERIAL NO.

VOLTS 120 FREQ. 60

NO. CKTS. NO. OF STEPS

STEPPING CONTACT SYMBOL "02" . . .

ROTARY SWITCH SYMBOL "05" — 12 POINT

16 POINT

ENCLOSURE NEMA 1 . . . .

BULLETIN 910C-1
CIRCUIT DESIGN CHART
FOR BULLETIN 910 STEP SWITCH

<u>3MT</u>

PREFILL

SYSTEM OPERATION	CONTROL DEVICE TO ADVANCE TO NEXT STEP	STEP	X	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
STAND BY	SM-J	1	X																			
"READY"	INTERLOCKS	2	X					X	X				X									
PREFILL OPEN	RDP-RO-O	3	X	X							X	X										
"	RPS-RO-O	4	X	X							X	X										
"	RDP-PRE-O	5	X	X	X						X	X										
"	RPS-PRE-O	6	X	X	X						X	X										
"	RPS-PRE-D	7	X	X	X						X	X										
"	RPS-PRE-D	8	X	X	X	X					X	X										
MANUAL HOLD	SM-J	9	X	X	X	X				X	X											
PREFILL CLOSED	RDP-RO-C	10	X		X						X											
"	RPS-RO-C	11	X		X						X											
"	RDP-PRE-C	12	X		X						X											
"	RPS-PRE-C	13	X				X				X											
HOME TO POS. 1	3MT TAP SW.	14	X								X											
"		15	X								X											
"		16	X								X											
		LOADS		R2 RELAY	R3 "	PILOT LIGHT PREFILL	1 MT STEP SW.	3 MT INTERLOCK	2 MT "	3 MT STEP SW.	3 MT HOMING	3 MT "	4 MT INTERLOCK									

LOAD SWITCHES ON STEP SWITCH — CLOSED: X OPEN: O

TYPE NO. MT

SERIAL NO.

VOLTS 120 FREQ. 60

NO. CKTS. ______ NO. OF STEPS ______

STEPPING CONTACT SYMBOL "02" . . . ______

ROTARY SWITCH SYMBOL "05" — 12 POINT ______

16 POINT ______

ENCLOSURE NEMA 1 . . . . ______

BULLETIN 910C-1
CIRCUIT DESIGN CHART
FOR BULLETIN 910 STEP SWITCH

4MT

JACK RETRACTION

SYSTEM OPERATION	CONTROL DEVICE TO ADVANCE TO NEXT STEP	STEP	Z	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
STAND BY	SM-J	1	X																			
"READY"	INTERLOCKS	2	X	X							X	X										
JACK RETRACT	DP-RI-O	3	X	X									X	X								
"	PS-RI-O	4	X	X									X	X								
"	RPS-RI-D	5	X	X									X	X								
"	RPB-RI-D	6	X	X	X	X	X	X	X				X	X								
"	RLS-1	7	X	X	X	X	X	X	X	X			X	X								
MANUAL HOLD	SM-J	8	X	X	X	X	X	X	X				X									
HOME TO POS. 1		9	X										X									
"		10	X										X									
"		11	X										X									
"		12	X										X									
		13																				
		14																				
		15																				
		16																				

LOAD SWITCHES ON STEP SWITCH — CLOSED: X OPEN: O

LOADS:
- R6 RELAY
- R12 " TOP
- R12 " BOTTOM
- R19 " ACC. 2
- R20 " ACC. 3
- PILOT LT.-JACK. O.
- 1 MT STEP SW.
- 2 MT INTERLOCK
- 3 MT "
- 4 MT HOMING
- 4 MT "

TYPE NO. MT ____________

SERIAL NO. ____________

VOLTS 120 FREQ. 60

NO. CKTS. ________ NO. OF STEPS ________

STEPPING CONTACT SYMBOL "02" ________

ROTARY SWITCH SYMBOL "05" — 12 POINT ________

16 POINT ________

ENCLOSURE NEMA 1 ________

DETAILED EXPLANATION OF COUNT RELAYS

AND HOW THEY CONTROL BOTH EJECTORS

AND ACCUMULATORS TWO AND THREE.

COUNTING

1 COUNT:

Count selector switch is turned to 1 count. "1 hit" pilot light is energized. Thereafter:

1) Start button is pressed, energizing relay R-101 and 1 MT step switch

   a) Contact R-101-1 closes.
   b) Step switch 1MT advances to position 2.
   c) Contact 1MT-7 closes, maintaining relay R-101 circuit, in series with contact R-101-1.
   d) "1 cycle" pilot light is energized, indicating first cycle is in progress.

   (Note: Start button contact must be closed for period longer than 1/10 second).

2) Process takes place, cutting in step switches 2MT, 3MT and 4MT, as programmed, afterwhich 1MT completes its cycle, and self steps to position 1.

   a) Contact 1MT-7 opens in position 1, acting as a "stop" button, and dropping out relay R-101.
   b) 1 cycle pilot light is deenergized.

   Single cycle is concluded.

2 COUNTS:

Count selector switch is turned to 2 counts. "2 hit" pilot light is energized. Thereafter:

3) Starting is accomplished as described in paragraph 2) above.

4) Process takes place, cutting in step switches 2MT, 3MT and 4MT, as programmed, afterwhich 1 MT completes its cycle, and self steps to position 1.

5) In position 12, contact 1MT-8 closes, energizing relay R-102 and "2 cycles" pilot light.

   a) Contact R-102-1 opens, deenergizing 1 cycle pilot.
   b) Contact R-102-2 closes, placing this contact in parallel with contact 1MT-7.
   c) Contact R-102-3 closes, maintaining relay R-102 circuit.
   d) Contact R-102-4 closes, setting up relay R-104 for later energization.

2 COUNTS: (Cont'd)

6) On arriving in position 1, even though contact 1MT-7 opens, relay R-101 does not drop out, as contact R-102-2 shunts open contact 1MT-7.

7) Step switch 1MT immediately steps to position 2.

8) Second cycle process takes place, cutting in step switches 2MT, 3MT and 4MT, as programmed, afterwhich 1MT completes its cycle, and self steps to position 1.

9) In position 10, contact 1MT-10 closed, energizing relay R-104:

   a) Contact R-104-1 opens, removing shunting effect of contact R-102-2 around contact 1MT-7.
   b) Contact R-104-2 closes, maintaining relay R-104 circuit.

10) Now opening of contact 1MT-7 in position 1, acts as "stop" button and drops out relay R-101.

   Contact R-101-1 opens, also deenergizing relays R-102 and R-104, and 2 cycles pilot light.

   All holding circuits are removed, and process is in its deenergized position.

   Two cycles are concluded.

3 COUNTS:

Count selector switch is turned to 3 counts, closing different contacts on various lines. "3 hit" pilot light is energized. Thereafter:

11) Starting is accomplished as described in paragraph 2) above

12) Process takes place, cutting in step switches 2MT, 3MT and 4MT, as programmed, afterwhich 1MT completes its cycle, and self steps to position 1.

13) In position 12, contact 1MT-8 closes, energizing relay R-102 and "2 cycles" pilot light:

   a) Contact R-102-1 opens, deenergizing 1 cycle pilot.
   b) Contact R-102-2 closes, placing this contact in parallel with contact 1MT-7.
   c) Contact R-102-3 closes, maintaining relay R-102 circuit.
   d) Contact R-102-4 closes, but has no effect in the circuit.

3 COUNTS: (Cont'd)

14) On arriving in position 1, even though contact 1MT-7 opens, relay R-101 does not drop out, as contact R-102-2 shunts open contact 1MT-7.

15) Step switch 1MT immediately steps to position 2.

16) Second cycle process takes place, cutting in step switches 2MT, 3MT and 4MT, as programmed, afterwhich 1MT completes its cycle, and self steps to position 1.

17) In position 11, contact 1MT-9 closes, energizing relay R-103 and "3 cycles" pilot light, indicating 3rd cycle in progress:

   a) Contact R-103-1 closes, placing this contact in parallel with contact 1MT-7.
   b) Contact R-103-2 opens, deenergizing 2 cycle pilot.
   c) Contact R-103-3 closes, maintaining relay R-103 circuit.
   d) Contact R-103-4 closes, setting up relay R-104 for later energization.

18) On arriving in position 1, even though contact 1MT-7 opens, relay R-101 does not drop out, as contacts R-102-2 and R-103-1 shunts open contact 1MT-7.

19) Step switch 1MT immediately steps to position 2.

20) Third cycle process takes place, cutting in step switches 2MT, 3MT and 4MT, as programmed, afterwhich 1MT completes its cycle, and self steps to position 1.

21) In position 10, contact 1MT-10 closed, energizing relay R-104.

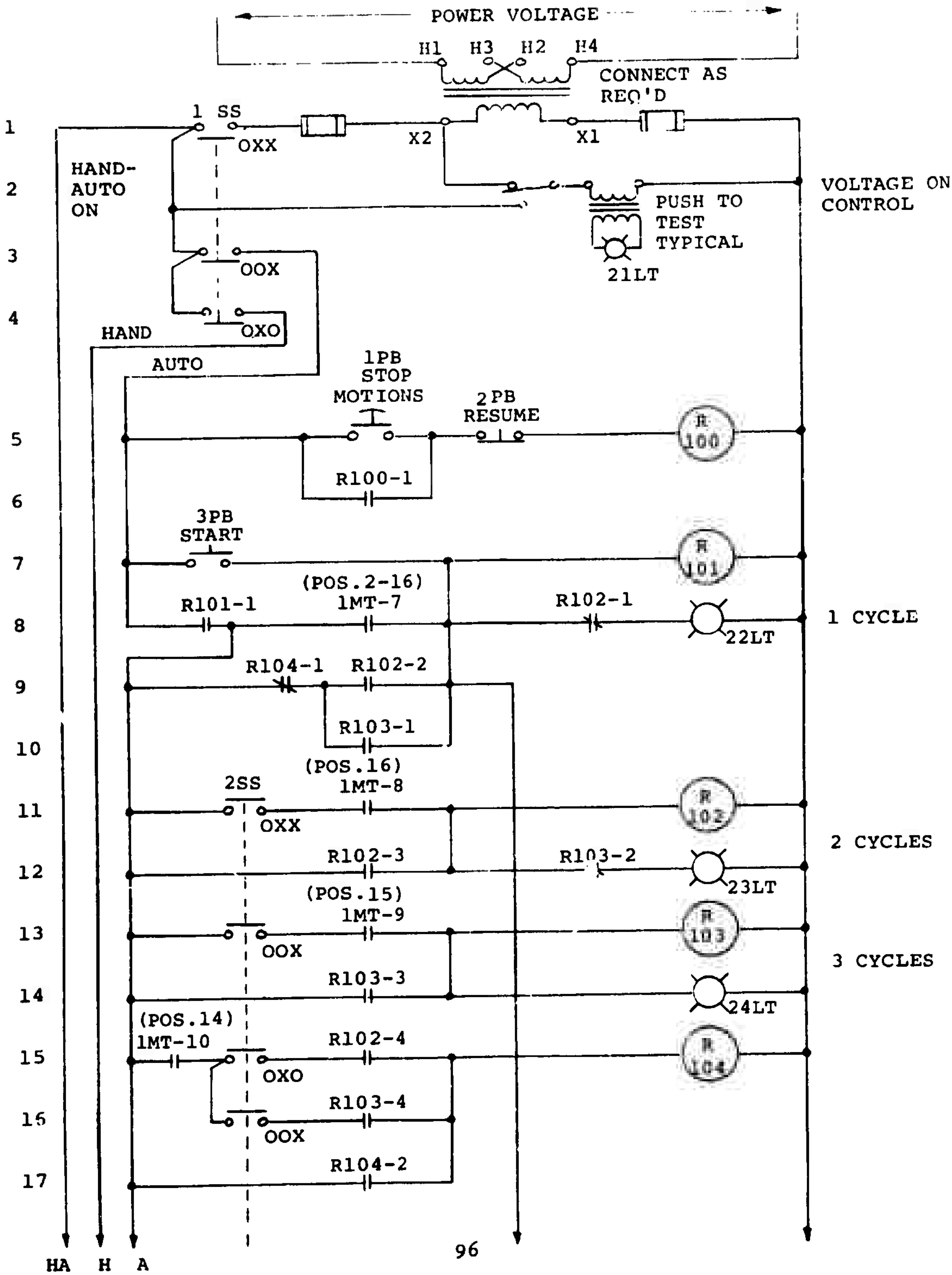
   a) Contact R-104-1 opens, removing shunting effect of contact R-102-2 around contact 1MT-7.
   b) Contact R-104-2 closes, maintaining relay R-104 circuit.

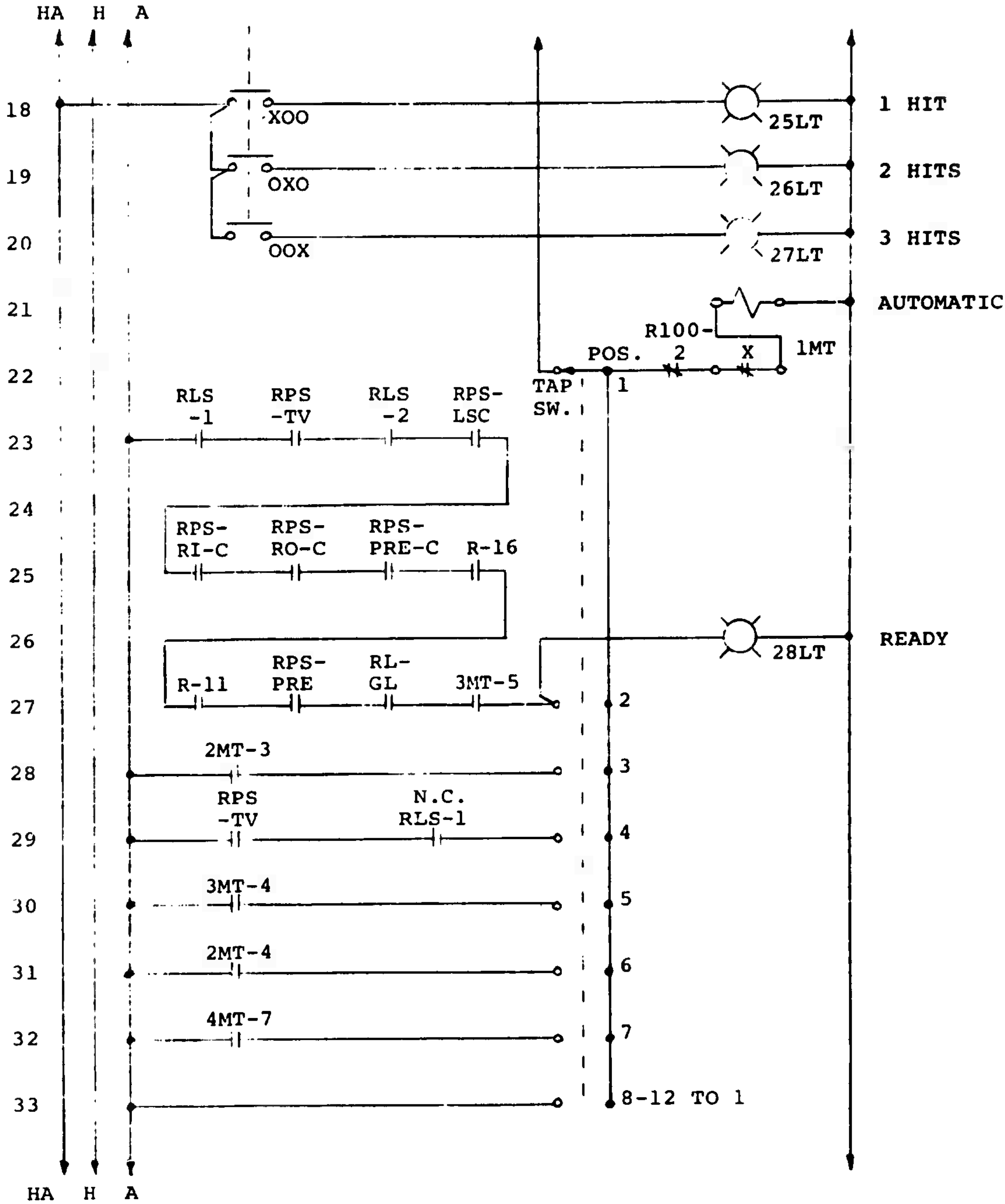
22) Now opening of contact 1MT-7 in position 1, acts as "stop" button and drops out relay R-101.

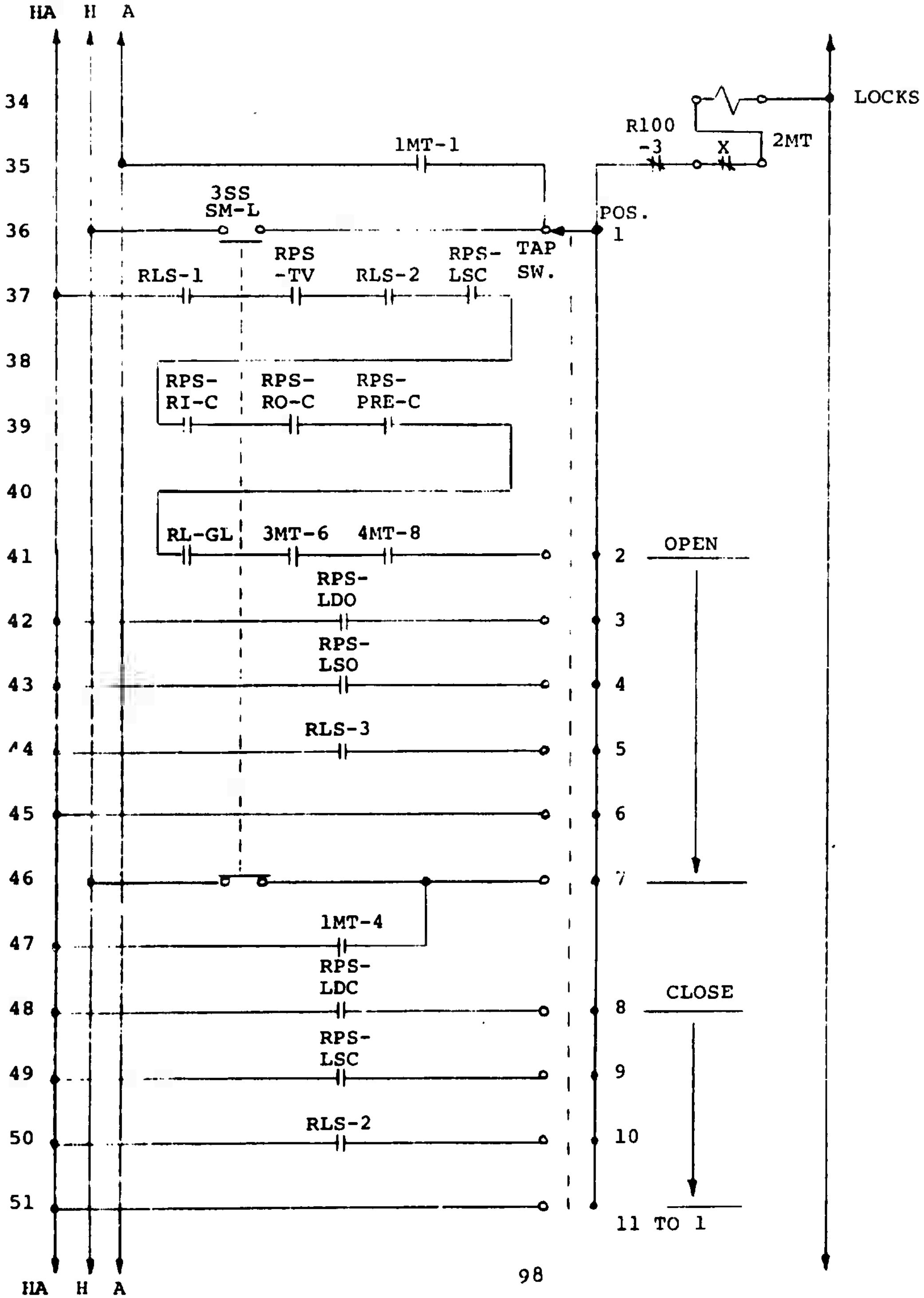
    Contact R-101-1 opens, also deenergizing relays R-102, R-103 and R104 and 3 cycles pilot light.

    All holding circuits are removed, and process is in its deenergized position.

    Three cycles are concluded.


POWER VOLTAGE
H1 H3 H2 H4
CONNECT AS REQ'D
1 SS
OXX
X2
X1
HAND-AUTO ON
VOLTAGE ON CONTROL
PUSH TO TEST TYPICAL
21LT
OOX
OXO
HAND
AUTO
1PB STOP MOTIONS
2PB RESUME
R100
R100-1
3PB START
R101
(POS.2-16) 1MT-7
R101-1
R102-1
22LT
1 CYCLE
R104-1
R102-2
R103-1
(POS.16) 1MT-8
2SS
OXX
R102
R102-3
R103-2
23LT
2 CYCLES
(POS.15) 1MT-9
OOX
R103
R103-3
24LT
3 CYCLES
(POS.14) 1MT-10
R102-4
OXO
R104
R103-4
OOX
R104-2
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
HA
H
A


HA H A
18
19
20
21
22
23
24
25
26
27
28
29
30
31
32
33
XOO
OXO
OOX
25LT
26LT
27LT
1 HIT
2 HITS
3 HITS
AUTOMATIC
R100-
POS. 2 X
1MT
TAP SW.
1
RLS -1
RPS -TV
RLS -2
RPS- LSC
RPS- RI-C
RPS- RO-C
RPS- PRE-C
R-16
R-11
RPS- PRE
RL- GL
3MT-5
28LT
READY
2
2MT-3
3
RPS -TV
N.C. RLS-1
4
3MT-4
5
2MT-4
6
4MT-7
7
8-12 TO 1
HA H A


HA H A
LOCKS
R100 -3
X
2MT
1MT-1
3SS SM-L
POS. 1
TAP SW.
RLS-1
RPS -TV
RLS-2
RPS-LSC
RPS-RI-C
RPS-RO-C
RPS-PRE-C
RL-GL
3MT-6
4MT-8
OPEN
RPS-LDO
RPS-LSO
RLS-3
1MT-4
RPS-LDC
CLOSE
RPS-LSC
RLS-2
11 TO 1
34
35
36
37
38
39
40
41
42
43
44
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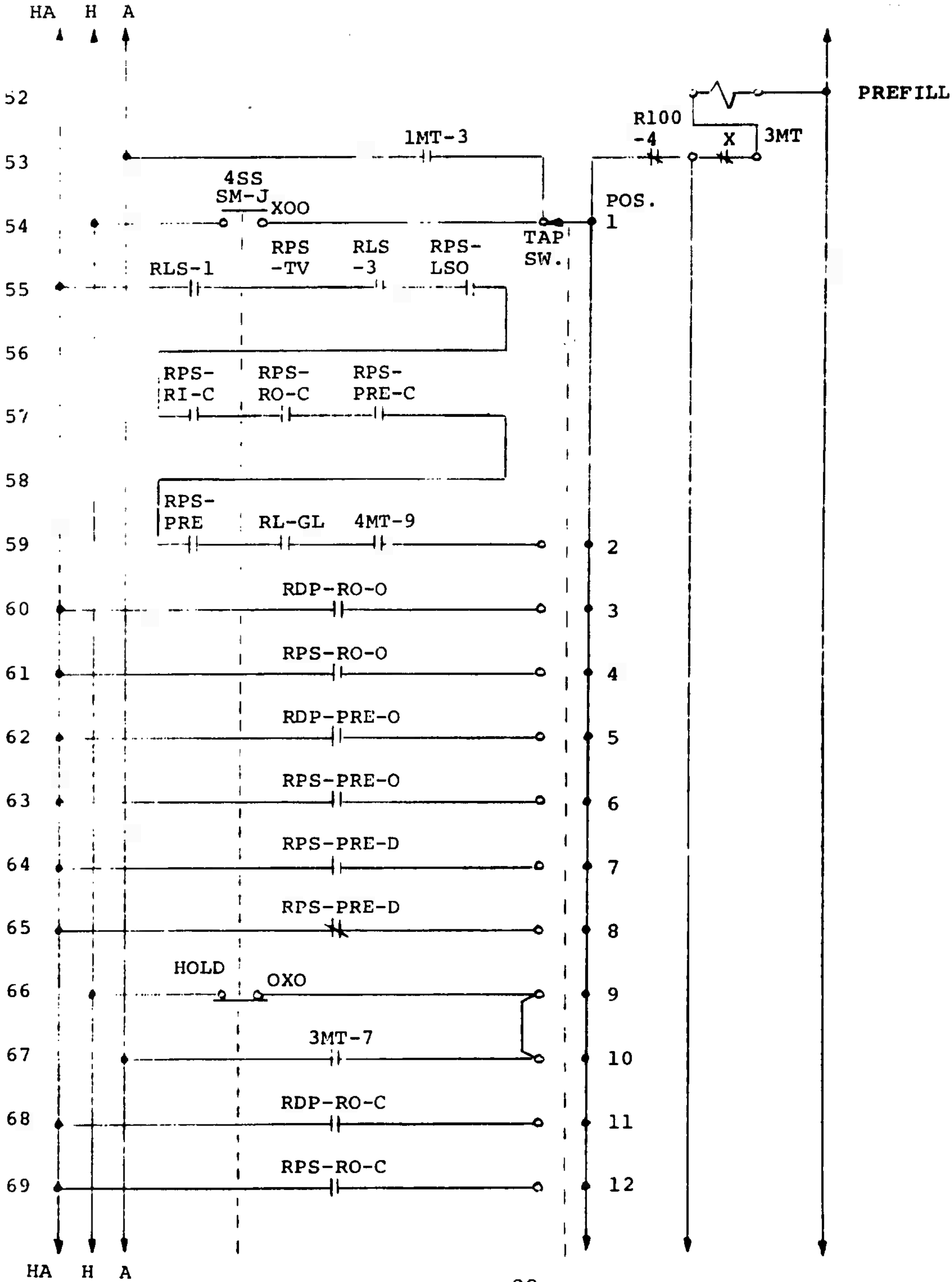
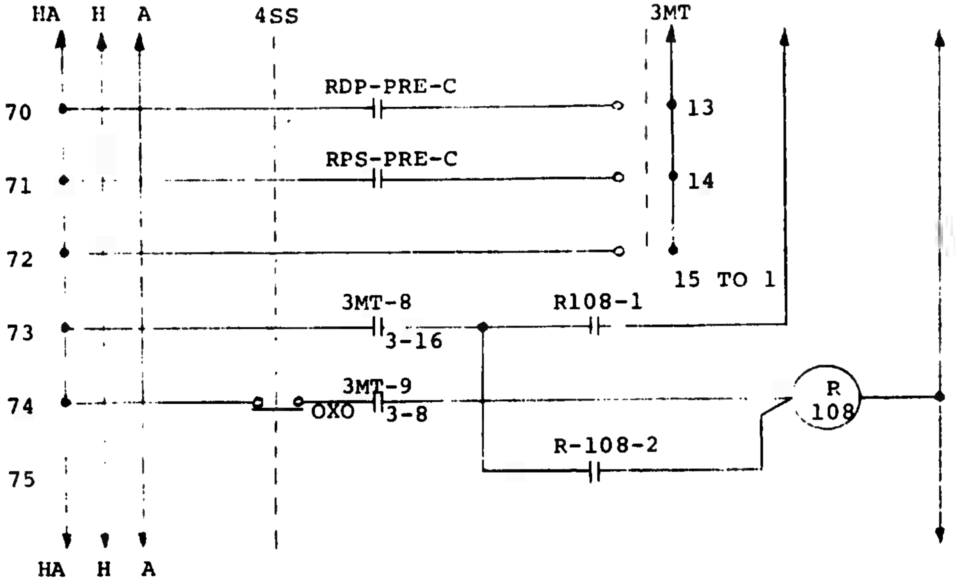
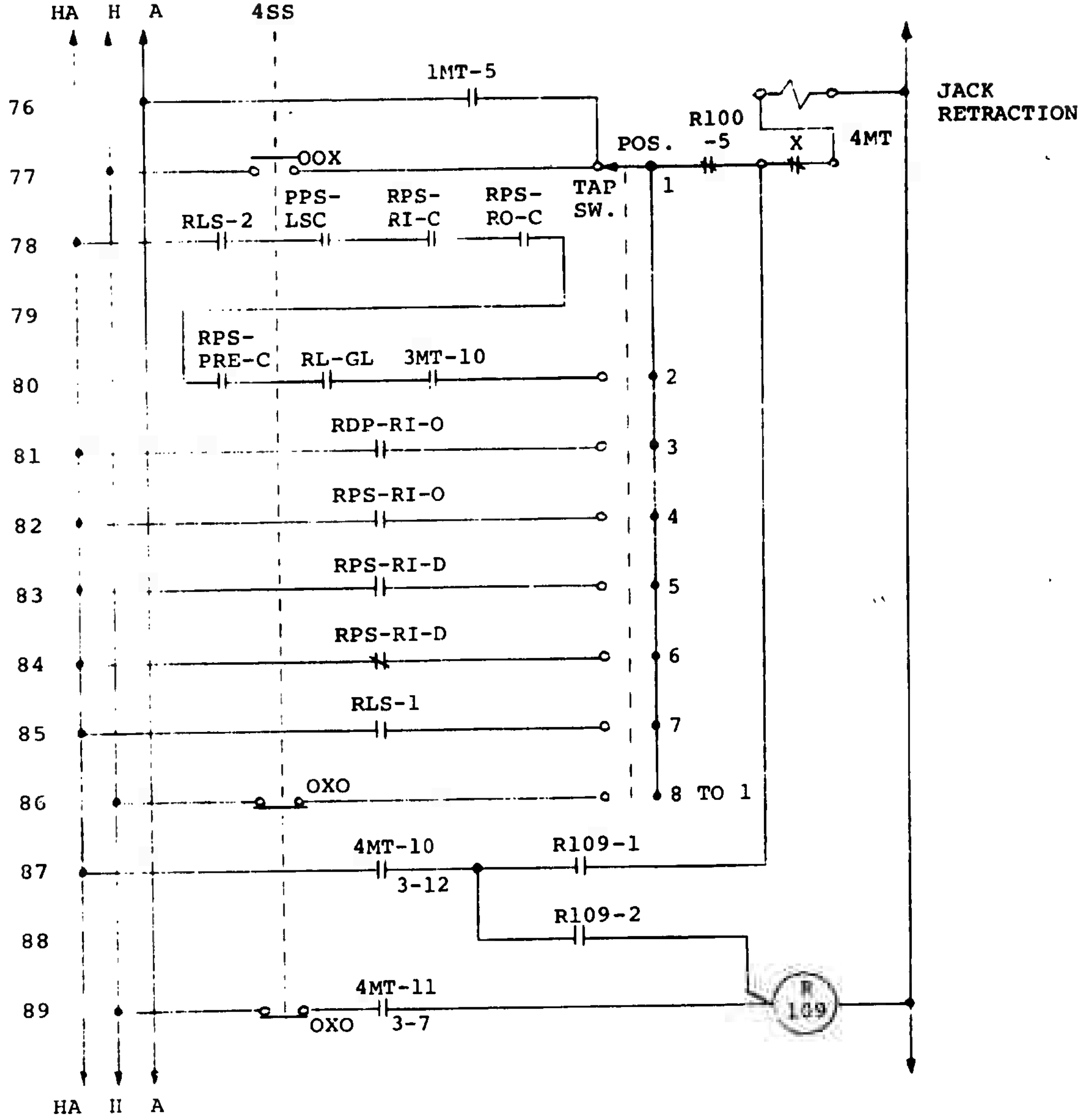
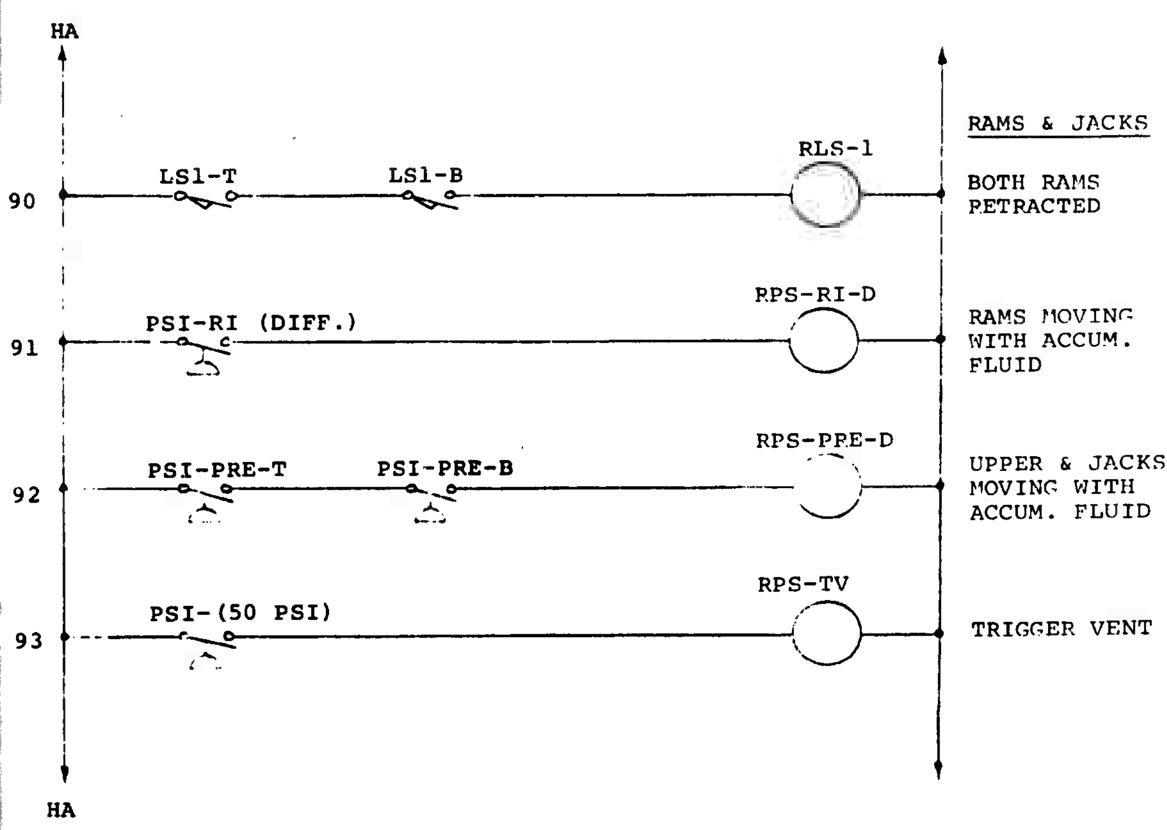
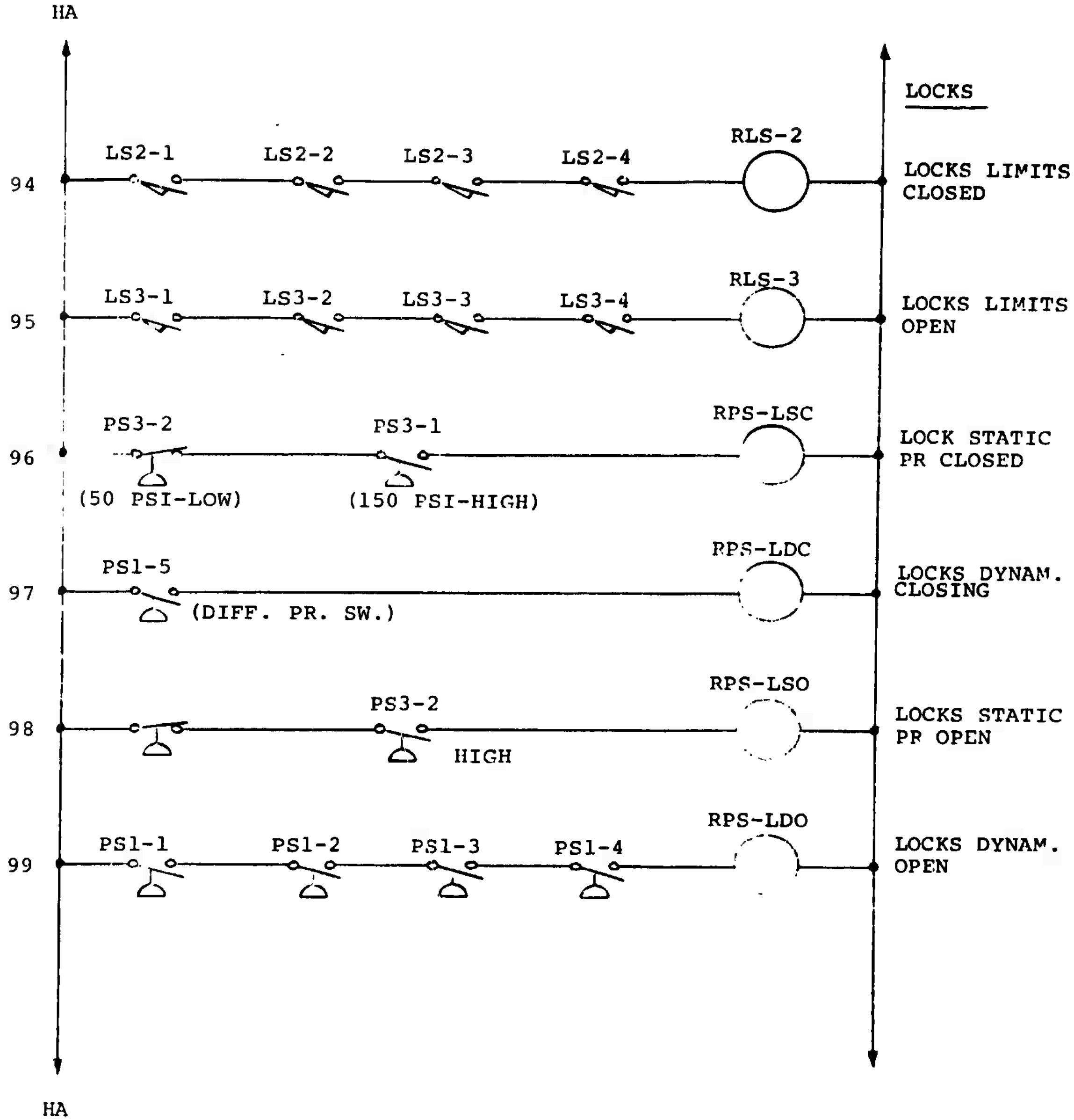
47
48
49
50
51
2
3
4
5
6
7
8
9
10
HA H A


HA
H
A
52
PREFILL
R100
-4
X
3MT
53
1MT-3
4SS
SM-J
XOO
POS.
1
54
TAP
SW.
RPS
-TV
RLS
-3
RPS-
LSO
55
RLS-1
56
RPS-
RI-C
RPS-
RO-C
RPS-
PRE-C
57
58
RPS-
PRE
RL-GL
4MT-9
59
2
60
RDP-RO-O
3
61
RPS-RO-O
4
62
RDP-PRE-O
5
63
RPS-PRE-O
6
64
RPS-PRE-D
7
65
RPS-PRE-D
8
66
HOLD
OXO
9
67
3MT-7
10
68
RDP-RO-C
11
69
RPS-RO-C
12
HA
H
A

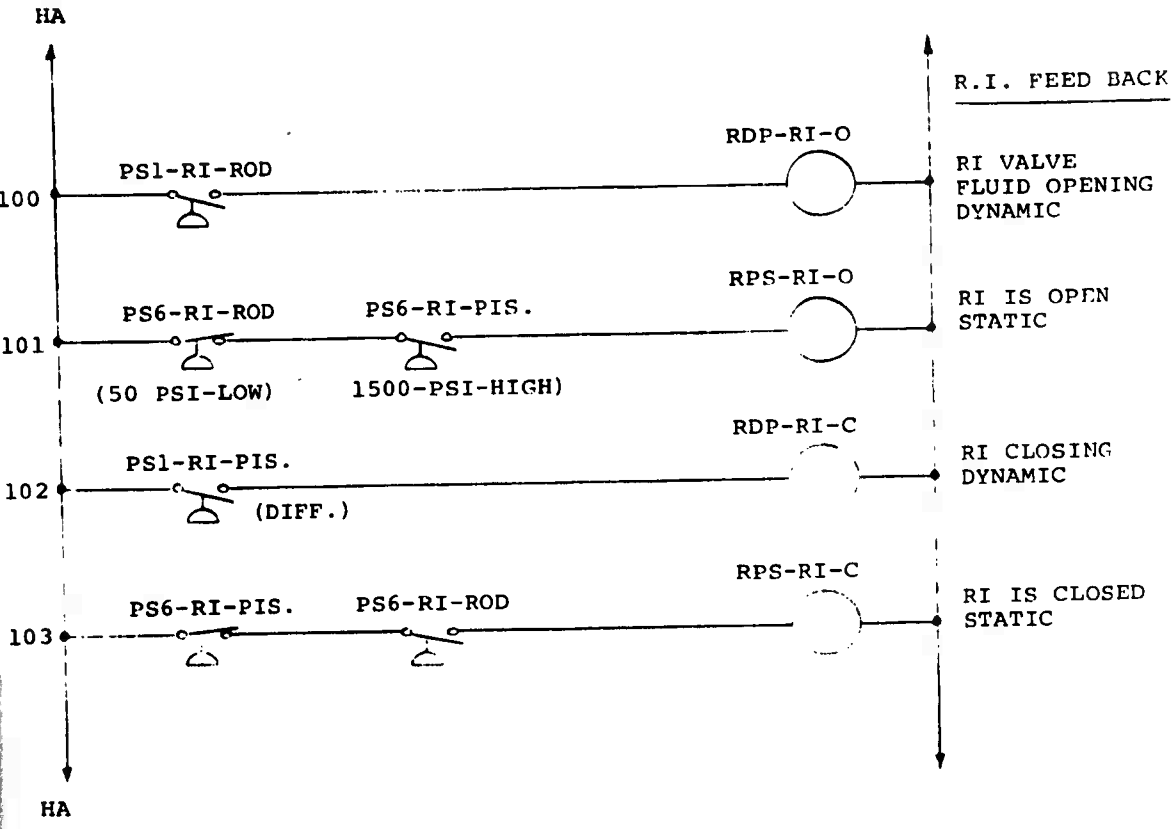

HA H A
4SS
3MT
RDP-PRE-C
13
RPS-PRE-C
14
15 TO 1
3MT-8
3-16
R108-1
3MT-9
OXO
3-8
R
108
R-108-2
70
71
72
73
74
75
HA H A

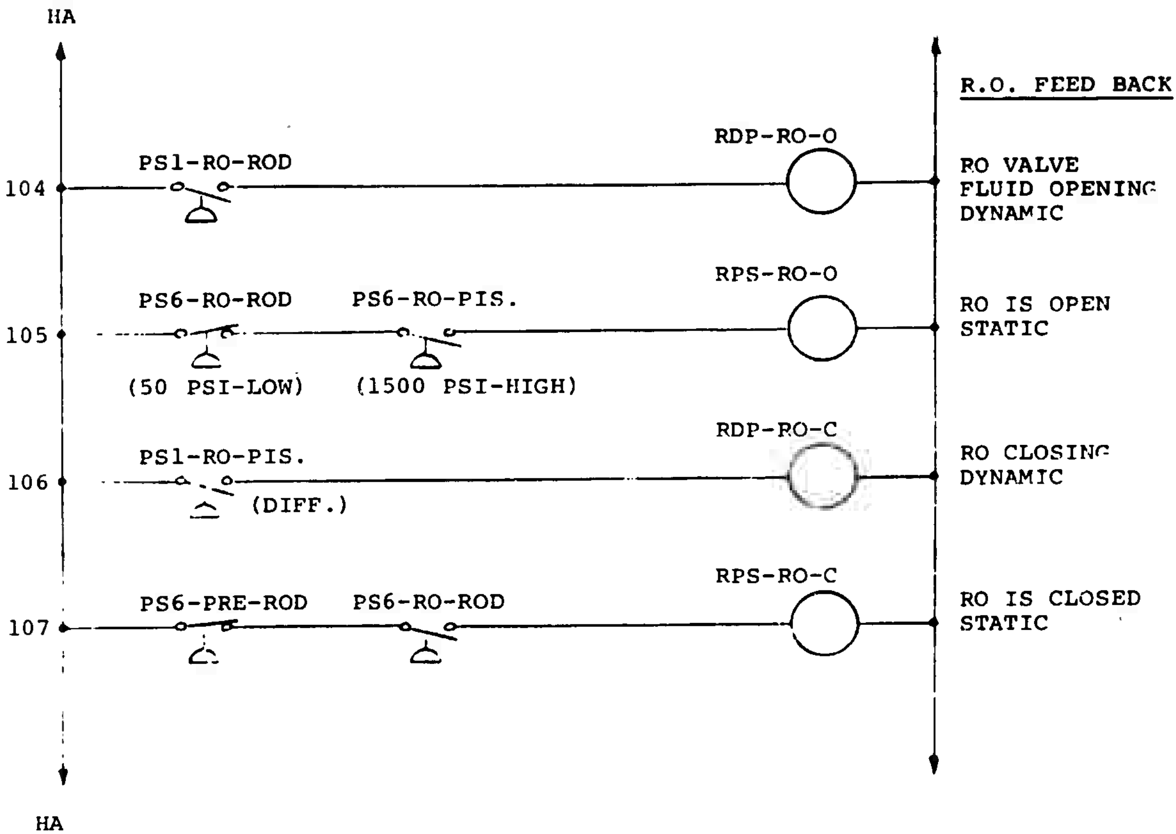

HA H A
4SS
1MT-5
JACK
RETRACTION
R100
-5
POS.
X
4MT
OOX
TAP
SW.
1
PPS-
LSC
RPS-
RI-C
RPS-
RO-C
RLS-2
RPS-
PRE-C
RL-GL
3MT-10
2
RDP-RI-O
3
RPS-RI-O
4
RPS-RI-D
5
RPS-RI-D
6
RLS-1
7
OXO
8 TO 1
4MT-10
3-12
R109-1
R109-2
4MT-11
3-7
OXO
R
109
76
77
78
79
80
81
82
83
84
85
86
87
88
89
HA II A


HA
RAMS & JACKS
RLS-1
90
LS1-T
LS1-B
BOTH RAMS RETRACTED
RPS-RI-D
91
PSI-RI (DIFF.)
RAMS MOVING WITH ACCUM. FLUID
RPS-PRE-D
92
PSI-PRE-T
PSI-PRE-B
UPPER & JACKS MOVING WITH ACCUM. FLUID
RPS-TV
93
PSI-(50 PSI)
TRIGGER VENT
HA

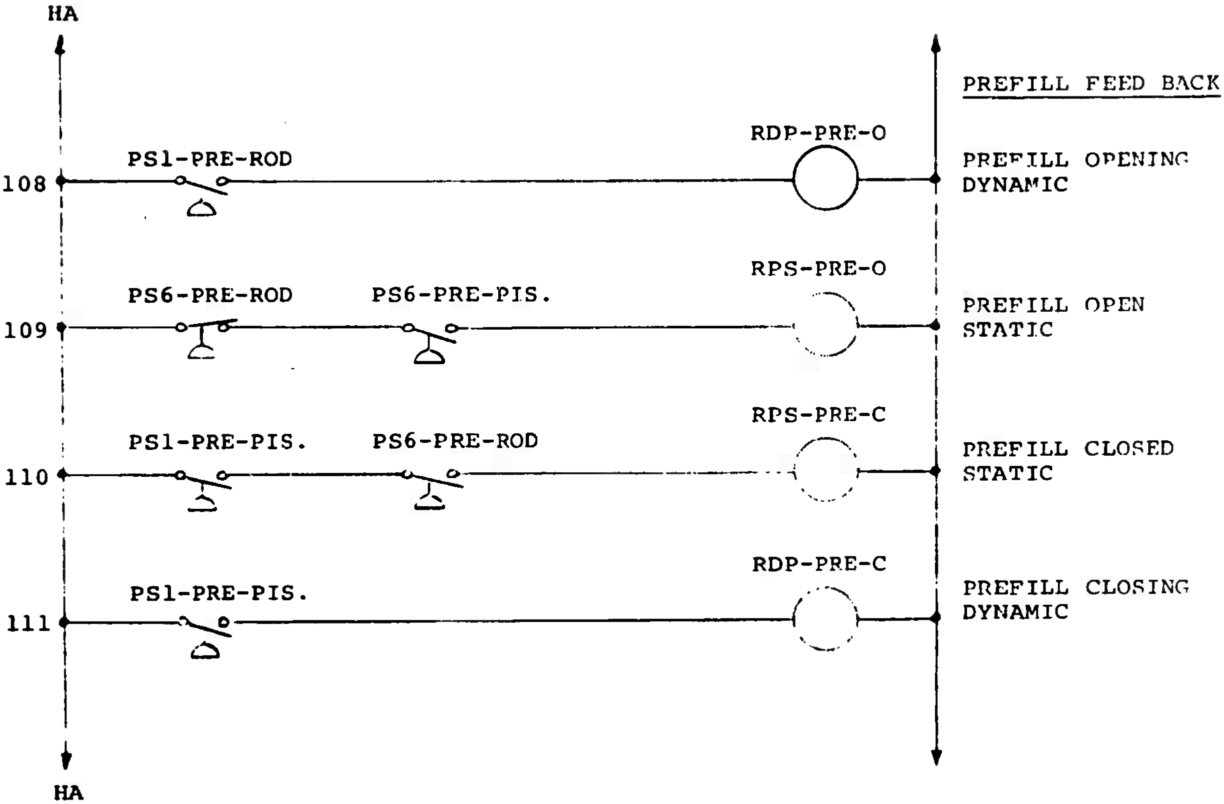


HA
LOCKS
LS2-1
LS2-2
LS2-3
LS2-4
RLS-2
94
LOCKS LIMITS CLOSED
LS3-1
LS3-2
LS3-3
LS3-4
RLS-3
95
LOCKS LIMITS OPEN
PS3-2
PS3-1
RPS-LSC
96
(50 PSI-LOW)
(150 PSI-HIGH)
LOCK STATIC PR CLOSED
PS1-5
RPS-LDC
97
(DIFF. PR. SW.)
LOCKS DYNAM. CLOSING
PS3-2
RPS-LSO
98
HIGH
LOCKS STATIC PR OPEN
PS1-1
PS1-2
PS1-3
PS1-4
RPS-LDO
99
LOCKS DYNAM. OPEN
HA


HA
R.I. FEED BACK
RDP-RI-O
PS1-RI-ROD
100
RI VALVE
FLUID OPENING
DYNAMIC
RPS-RI-O
PS6-RI-ROD
PS6-RI-PIS.
101
RI IS OPEN
STATIC
(50 PSI-LOW)
1500-PSI-HIGH)
RDP-RI-C
PS1-RI-PIS.
102
(DIFF.)
RI CLOSING
DYNAMIC
RPS-RI-C
PS6-RI-PIS.
PS6-RI-ROD
103
RI IS CLOSED
STATIC
HA

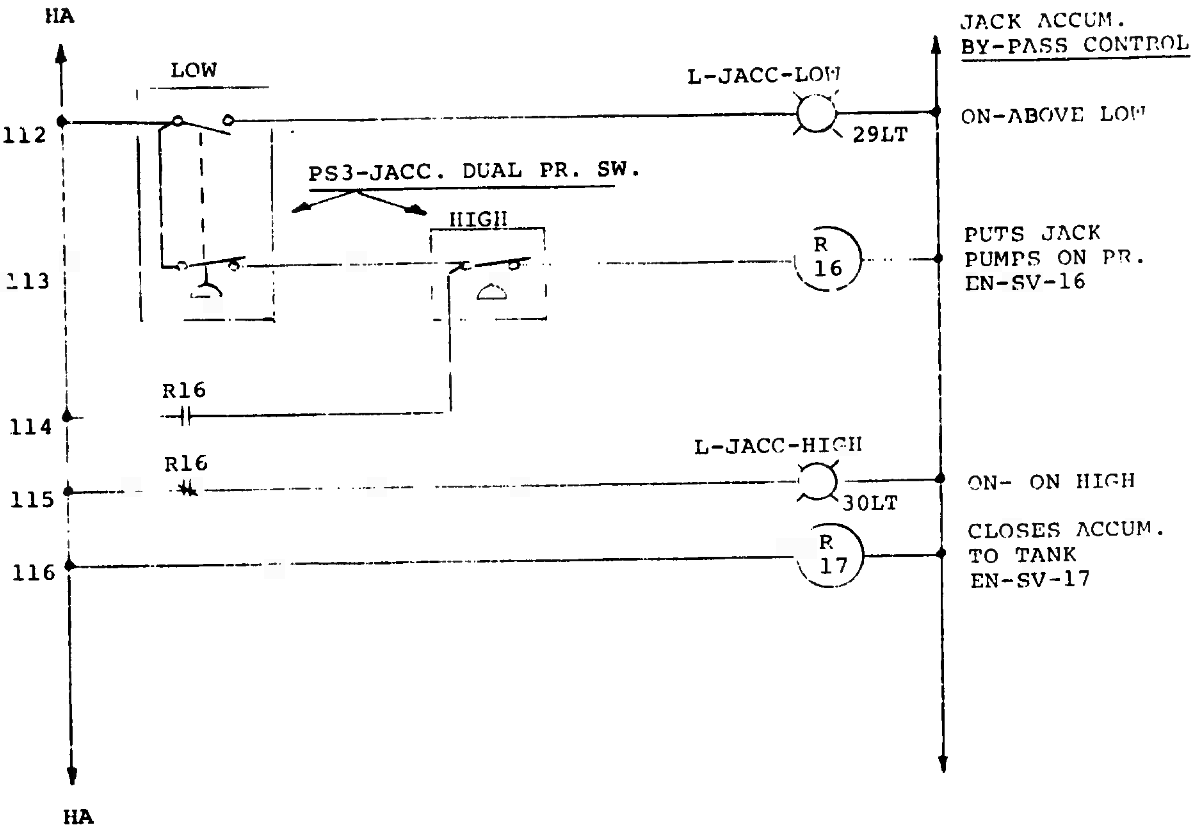


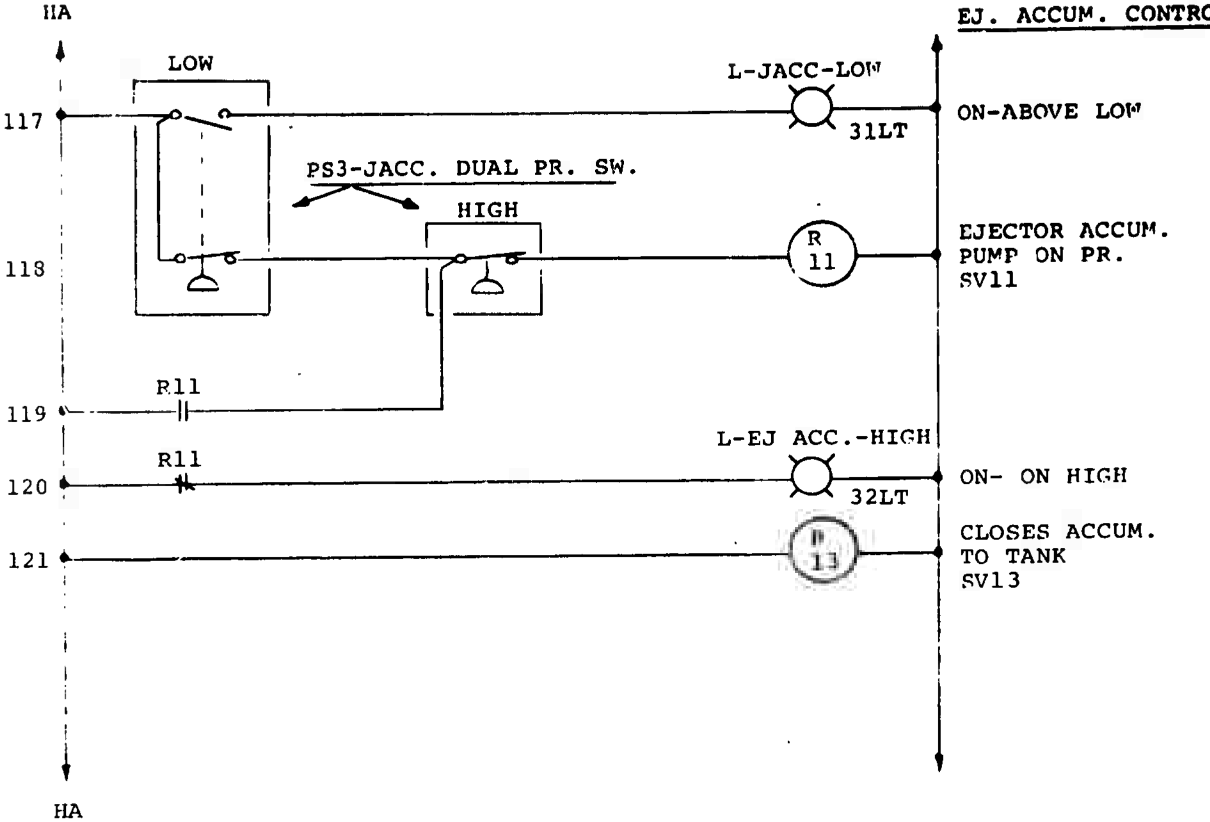
HA
R.O. FEED BACK
RDP-RO-O
PS1-RO-ROD
104
RO VALVE
FLUID OPENING
DYNAMIC
RPS-RO-O
PS6-RO-ROD
PS6-RO-PIS.
105
RO IS OPEN
STATIC
(50 PSI-LOW)
(1500 PSI-HIGH)
RDP-RO-C
PS1-RO-PIS.
106
RO CLOSING
DYNAMIC
(DIFF.)
RPS-RO-C
PS6-PRE-ROD
PS6-RO-ROD
107
RO IS CLOSED
STATIC
HA


HA
PREFILL FEED BACK
RDP-PRE-O
PS1-PRE-ROD
108
PREFILL OPENING DYNAMIC
RPS-PRE-O
PS6-PRE-ROD
PS6-PRE-PIS.
109
PREFILL OPEN STATIC
RPS-PRE-C
PS1-PRE-PIS.
PS6-PRE-ROD
110
PREFILL CLOSED STATIC
RDP-PRE-C
PS1-PRE-PIS.
111
PREFILL CLOSING DYNAMIC
HA

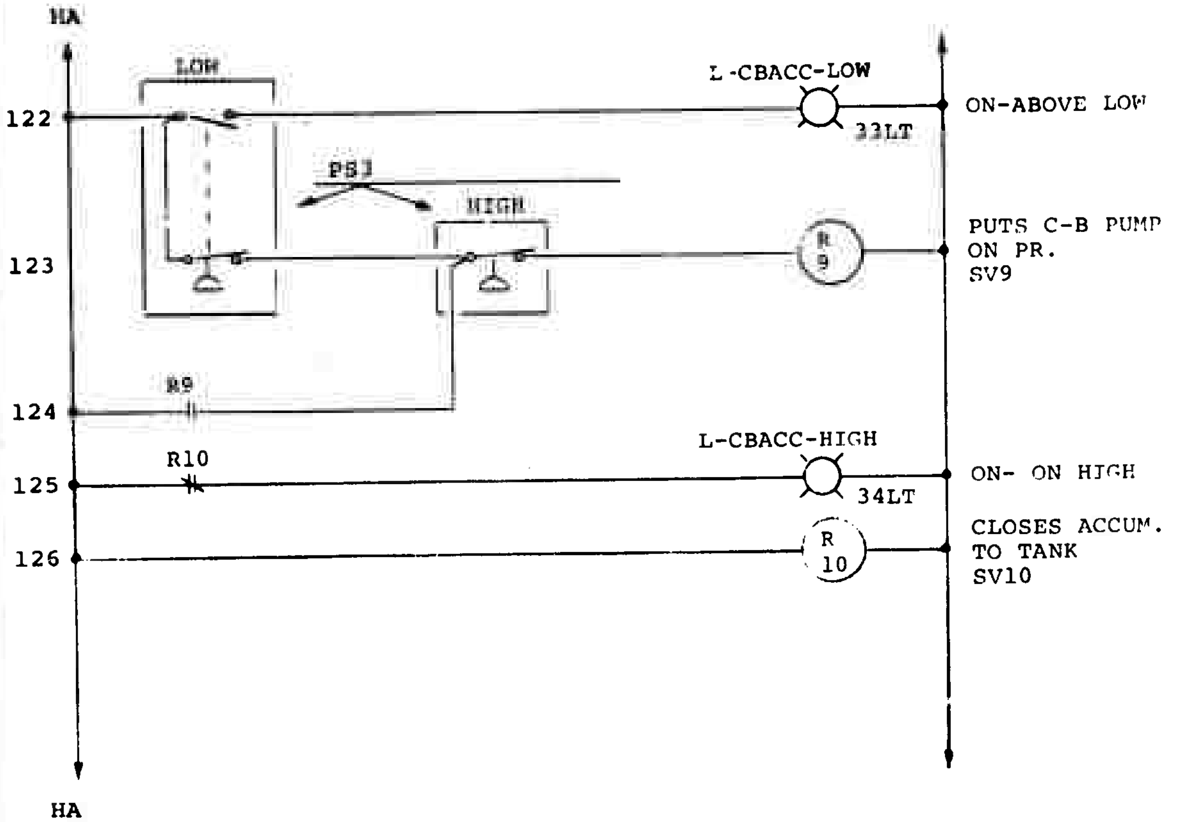
JACK, EJECTOR AND COUNTERBALANCE ACCUMULATOR BY-PASS CONTROLS

These three identical systems are controlled by a high-low pressure switch which puts the respective pump on Pr until the accumulator is charged. When the accumulator is charged, the high Pr switch contact opens, thereby de-energizing the by-pass relay and solenoid.


HA
JACK ACCUM.
BY-PASS CONTROL
LOW
L-JACC-LOW
112
ON-ABOVE LOW
29LT
PS3-JACC. DUAL PR. SW.
HIGH
R
16
PUTS JACK
PUMPS ON PR.
EN-SV-16
113
R16
114
L-JACC-HIGH
R16
115
ON- ON HIGH
30LT
R
17
CLOSES ACCUM.
TO TANK
EN-SV-17
116
HA

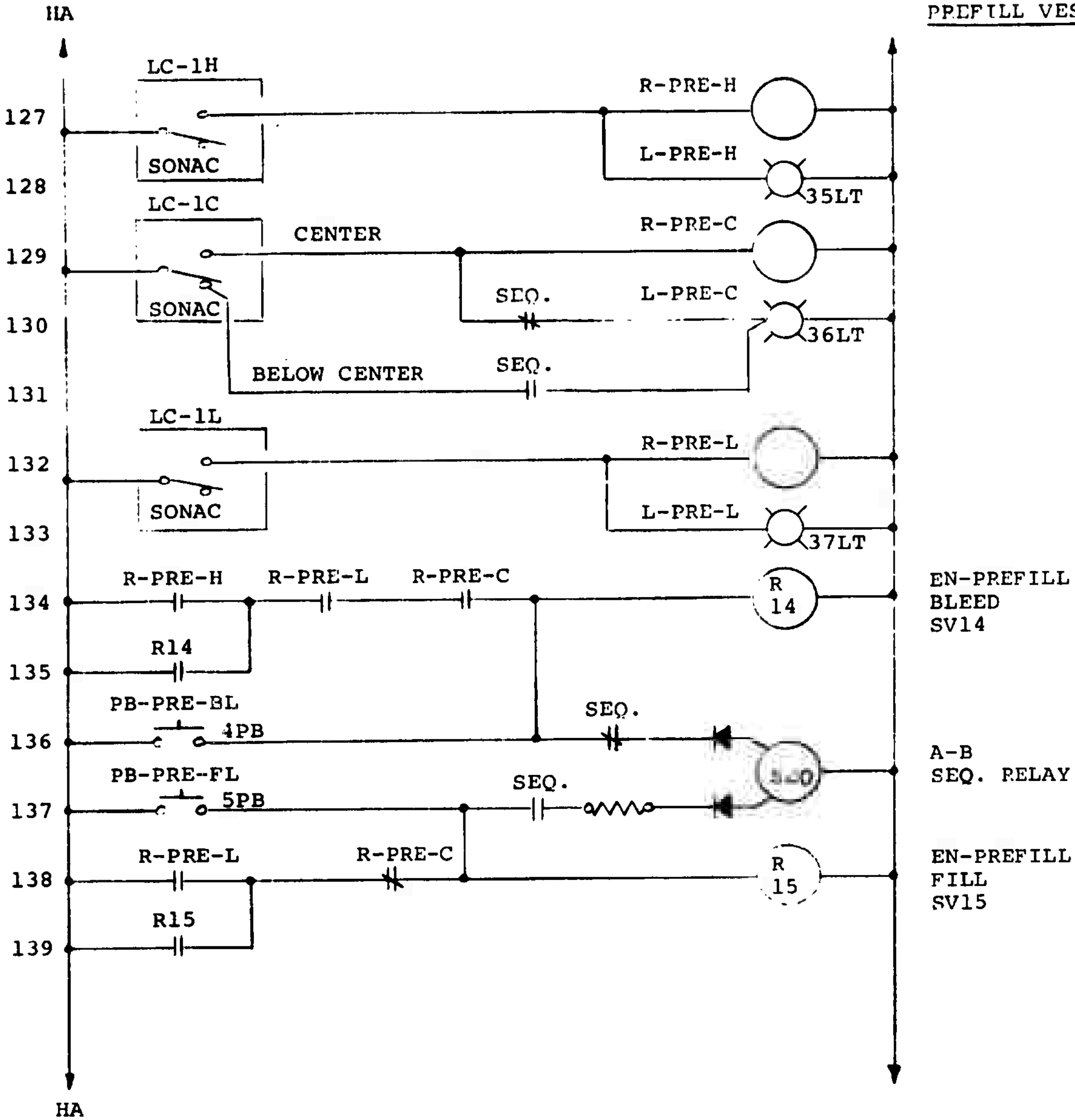


HA
EJ. ACCUM. CONTRO
LOW
L-JACC-LOW
117
31LT
ON-ABOVE LOW
PS3-JACC. DUAL PR. SW.
HIGH
118
R 11
EJECTOR ACCUM.
PUMP ON PR.
SV11
R11
119
L-EJ ACC.-HIGH
R11
120
32LT
ON- ON HIGH
121
R 13
CLOSES ACCUM.
TO TANK
SV13
HA


HA
LOW
L-CBACC-LOW
122
ON-ABOVE LOW
33LT
PS3
HIGH
PUTS C-B PUMP
ON PR.
SV9
123
R 9
124
R9
L-CBACC-HIGH
R10
125
ON- ON HIGH
34LT
126
R 10
CLOSES ACCUM.
TO TANK
SV10
HA

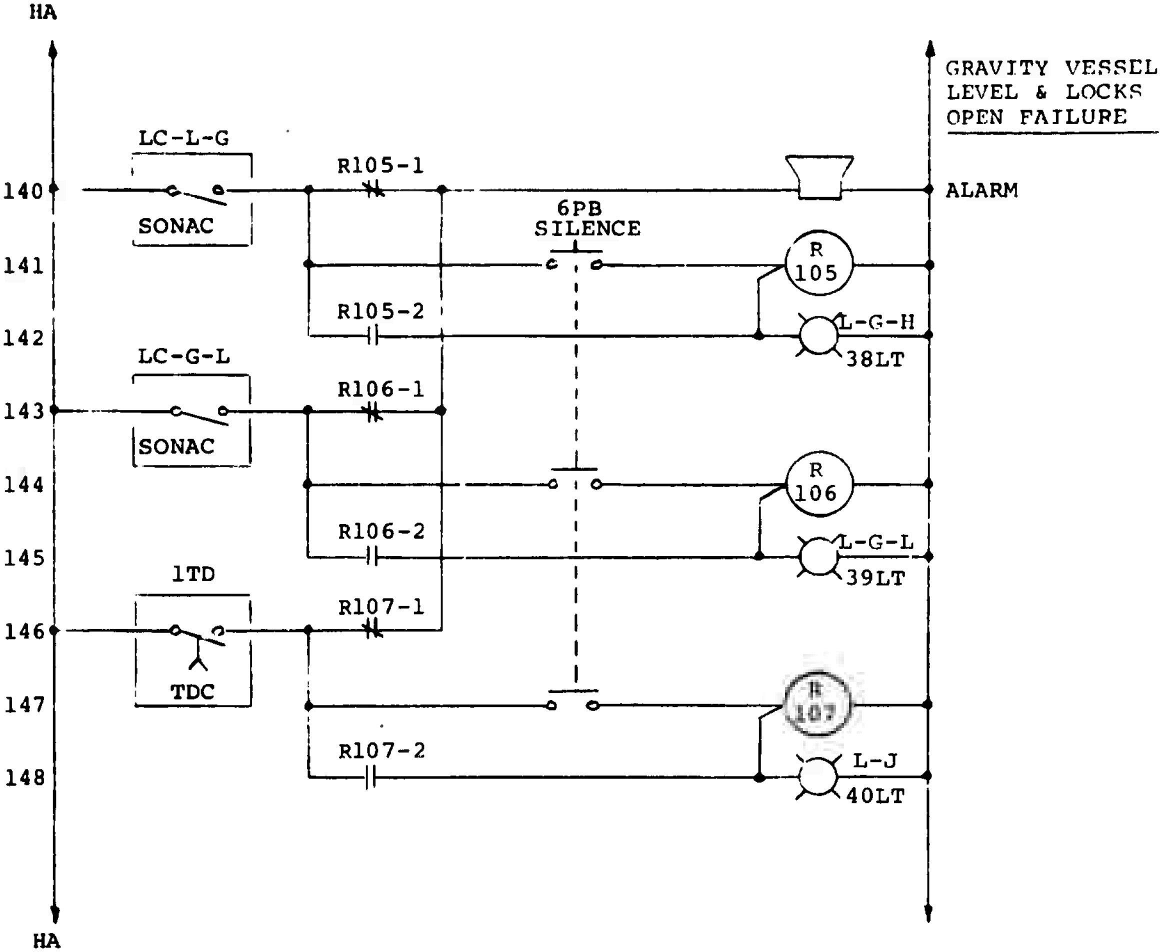
PREFILL VESSEL

The "Sonac" level control devices are located in the walls of the vessel in a vertical array. When the fluid level is low, relay R-PRE-L and light L-PRE-L are energized automatically. The energizing of this relay causes R-15 and SV-15 to be energized, filling the prefill vessel from the jack accumulator bank. When the fluid level reaches a point above the lowest Sonac, the lower light, L-PRE-L, goes out. The level contrinues to rise until the center Sonac is actuated, energizing the center light, L-PRE-C. If, for some reason, the level continues to rise, possibly due to leakage, the upper Sonac, LC-1-H, will be actuated. This Sonac will light L-PRE-H and energize relay R-PRE-H. When all three Sonacs are actuated, relay R-14 and its holding contact lock-in. This high level condition energizes the sequence relay which extinguishes the center light until the fluid level falls to the actuating level of the center Sonac, causing the center light to be illuminated again. Note that the level has dropped due to R-PRE-H and relay R-14 energizing the drain valve SV-14, causing the excess prefill fluid to empty into the gravity tank.


HA
LC-1H
R-PRE-H
127
SONAC
L-PRE-H
128
35LT
LC-1C
R-PRE-C
CENTER
129
SONAC
SEQ.
L-PRE-C
130
36LT
BELOW CENTER
SEQ.
131
LC-1L
R-PRE-L
132
SONAC
L-PRE-L
133
37LT
R-PRE-H
R-PRE-L
R-PRE-C
R 14
134
EN-PREFILL BLEED SV14
R14
135
PB-PRE-BL
4PB
SEQ.
136
A-B SEQ. RELAY
PB-PRE-FL
5PB
SEQ.
137
R-PRE-L
R-PRE-C
R 15
138
EN-PREFILL FILL SV15
R15
139
HA

GRAVITY VESSEL CONTROL & LOCKS OPEN ALARM

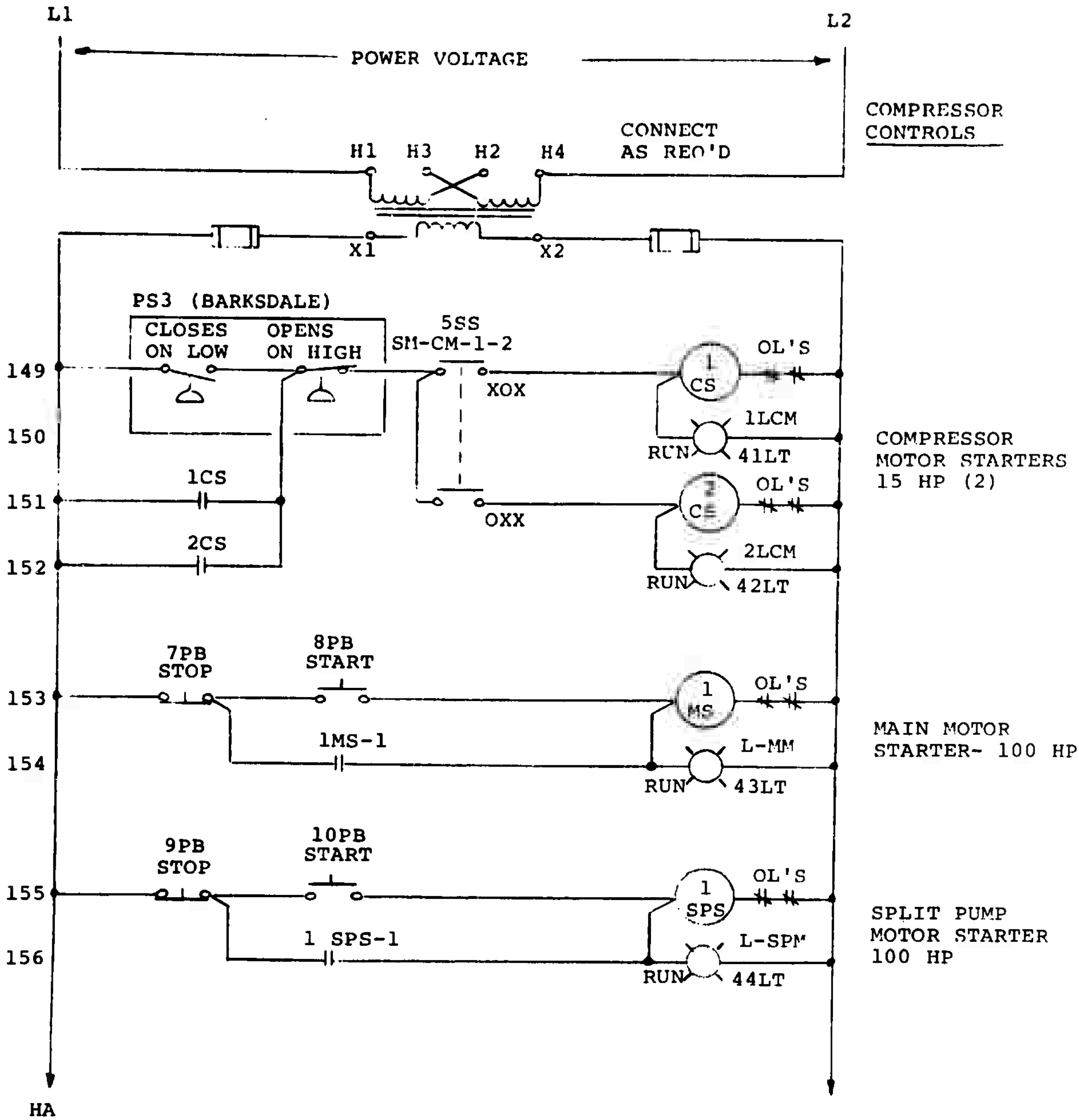
If either high or dangerously low fluid level occurs in the gravity tan', an audio alarm will sound. The same warning occurs if 1 TD times-out before the normal time required for lock opening. (See line 203.) Note that stepping contact 2 MT-2 corresponds to the normal sequence steps for opening the locks. Therefore, if 1 TD times-in while the 2 MT-2 is still closed, the alarm will sound. The operator can silence the alarm by depressing the silence push button. However, the relay system retains a memory effect for the malfunction and stores the malfunction data through relays R-105 and R-106 and their respective pilot lights.

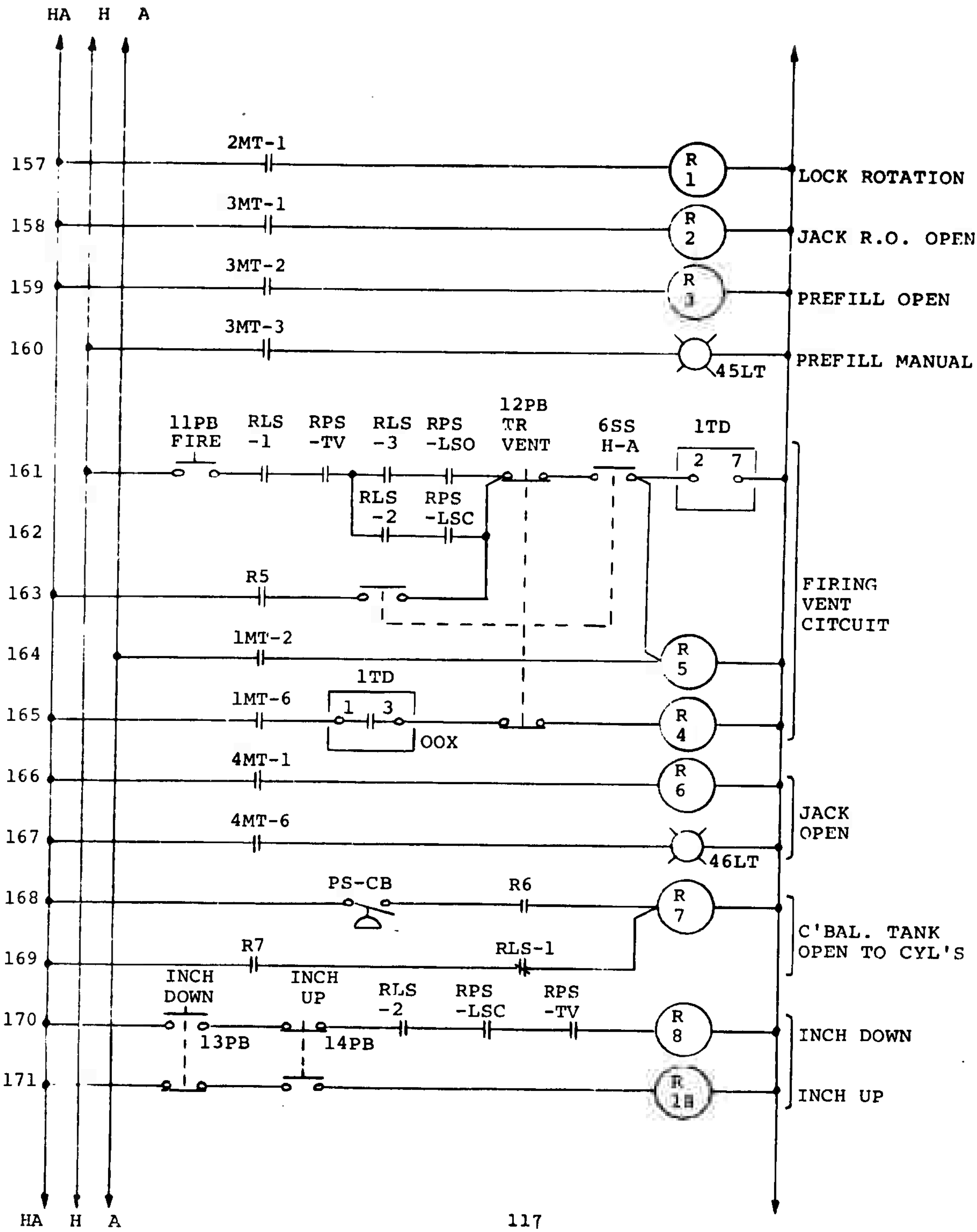


HA
GRAVITY VESSEL
LEVEL & LOCKS
OPEN FAILURE
LC-L-G
R105-1
140
SONAC
ALARM
6PB
SILENCE
141
R
105
R105-2
142
L-G-H
38LT
LC-G-L
R106-1
143
SONAC
144
R
106
R106-2
145
L-G-L
39LT
1TD
R107-1
146
TDC
147
R
107
R107-2
148
L-J
40LT
HA

COMPRESSOR & MOTOR CONTROLS

Both compressors are controlled by a dual pressure switch which connected to the high pressure receiver. The circuit is high-low enabling automatic cut-in and cut-out of the compressor depending on the setting of the Pr switch. The Pr switch must be mounted adjacent to the receiver Pr gauge on the console so that settings can be made easily. A four position selector, SM-CM-1-2, controls both compressor motors.


L1
L2
POWER VOLTAGE
COMPRESSOR CONTROLS
H1 H3 H2 H4
CONNECT AS REQ'D
X1
X2
PS3 (BARKSDALE)
CLOSES ON LOW
OPENS ON HIGH
5SS
SM-CM-1-2
XOX
OXX
149
150
151
152
1CS
2CS
1 CS
OL'S
1LCM
RUN
41LT
2 CS
OL'S
2LCM
RUN
42LT
COMPRESSOR MOTOR STARTERS 15 HP (2)
153
154
7PB STOP
8PB START
1MS-1
1 MS
OL'S
L-MM
RUN
43LT
MAIN MOTOR STARTER- 100 HP
155
156
9PB STOP
10PB START
1 SPS-1
1 SPS
OL'S
L-SPM
RUN
44LT
SPLIT PUMP MOTOR STARTER 100 HP
HA


HA H A
157 2MT-1 R 1 LOCK ROTATION
158 3MT-1 R 2 JACK R.O. OPEN
159 3MT-2 R PREFILL OPEN
160 3MT-3 45LT PREFILL MANUAL
161 11PB FIRE RLS -1 RPS -TV RLS -3 RPS -LSO 12PB TR VENT 6SS H-A 1TD 2 7
162 RLS -2 RPS -LSC
163 R5
164 1MT-2 R 5
165 1MT-6 1TD 1 3 OOX R 4
FIRING VENT CITCUIT
166 4MT-1 R 6
167 4MT-6 46LT
JACK OPEN
168 PS-CB R6 R 7
169 R7 RLS-1
C'BAL. TANK OPEN TO CYL'S
170 INCH DOWN 13PB INCH UP 14PB RLS -2 RPS -LSC RPS -TV R 8
171 R 18
INCH DOWN
INCH UP
HA H A

EJECTORS - TOP AND BOTTOM

CIRCUIT FEATURES:

a) Ejection may be omitted from automatic operation by selection. Ejection would, in this event, be accomplished by manual mode.

b) Ejection top and ejection bottom, are entirely independent operations. Either one or the other, or both, may be omitted.

c) This independence is extended to the difference in timing intervals that may be set for each one.

d) Both manual and automatic operations are programmed by step switch 4MT.

e) In automatic operation, ejection will take place

    After every single cycle.
    After the second (last) cycle ONLY of a selected two cycle operation.
    After the third (last) cycle ONLY of a selected three cycle operation.

f) The ejection sequence, as described in paragraph e) is a function of the setting of the count selector, and automatically produces the operation of last cycle ejection.

AUTOMATIC OPERATION - 1 HIT:

Off-On selector is turned to "on." Count selector switch is turned to 1 count. Thereafter:

1) At programmed point, contact 4MT-2 closes, energizing timer 1TR. This is a delayed interval.

2) Contact 4MT-2 closure may be sustained at this point, or may have open, but is effective either way, as alternate holding circuit has been set up in the timer.

3) On "time out" of Timer 1TR, 4 - 5 contact transfers to 4 - 3 to supply pulse start to timer 2TR, which is energized and maintained by its own holding circuit.

4) Relay R-12 (top) is energized, its contacts closing in turn, to energize Ejection Top solenoid valve. Ejection takes place.

5) When timer 2TR "times out," the ejector circuit is tripped to its deenergized position.

EJECTORS
2 HIT CYCLE
AUTOMATIC OPERATION:

Off-on selector is turned to "on." Count selector switch is turned to 2 cycles. Thereafter:

6) On first cycle, contact 4MT-2 closes, but produces no signal, as count selector switch is open in this position. Even though the count switch second contact is closed, still no signal to eject is produced, as contact R-102 is open.

7) On the 2nd cycle, and in preparation thereto, contact R-102 is now closed. When contact 4MT-2 closes, operation proceeds as described in paragraphs 2) through 5) above.

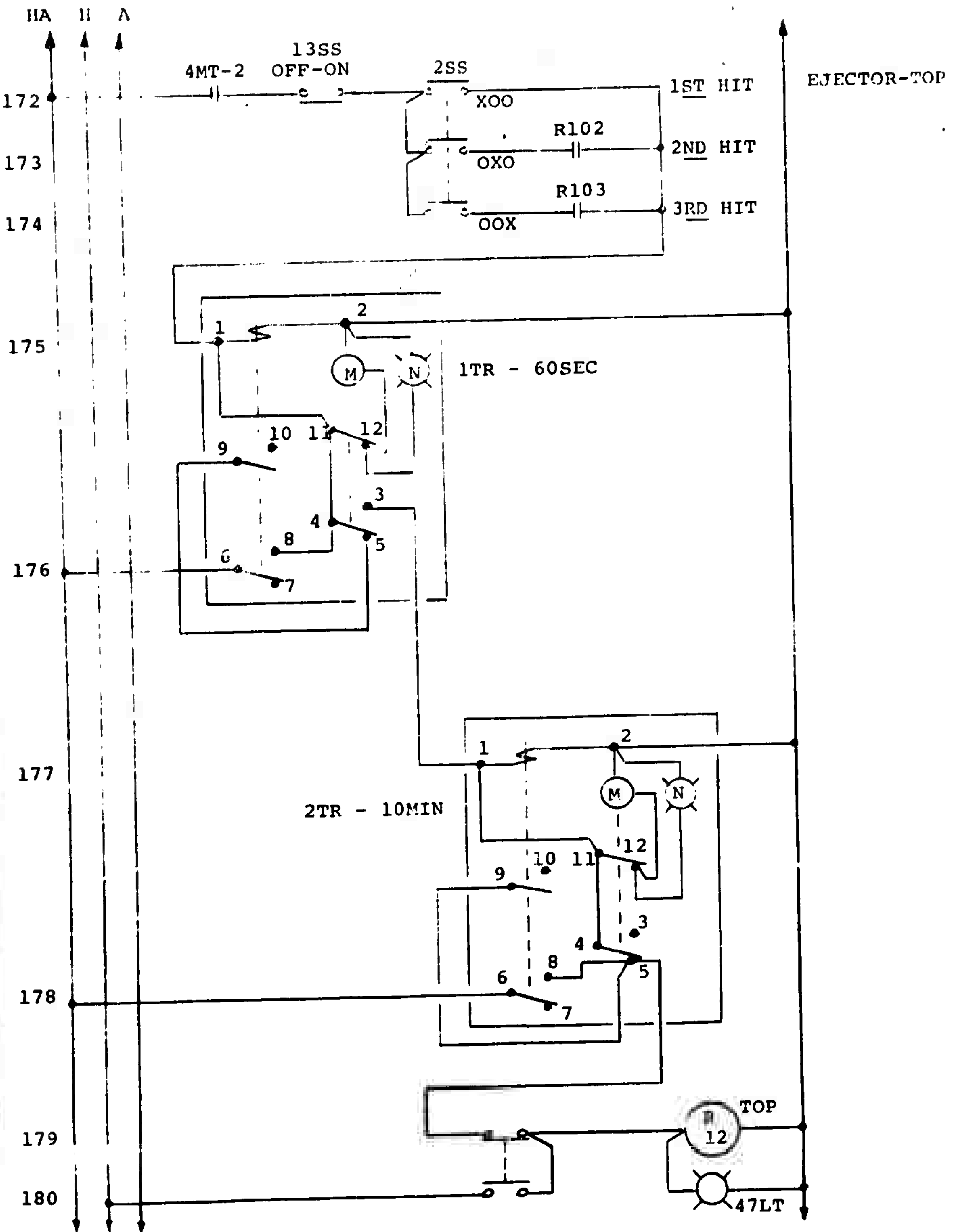
EJECTORS
3 HIT CYCLE
AUTOMATIC OPERATION:

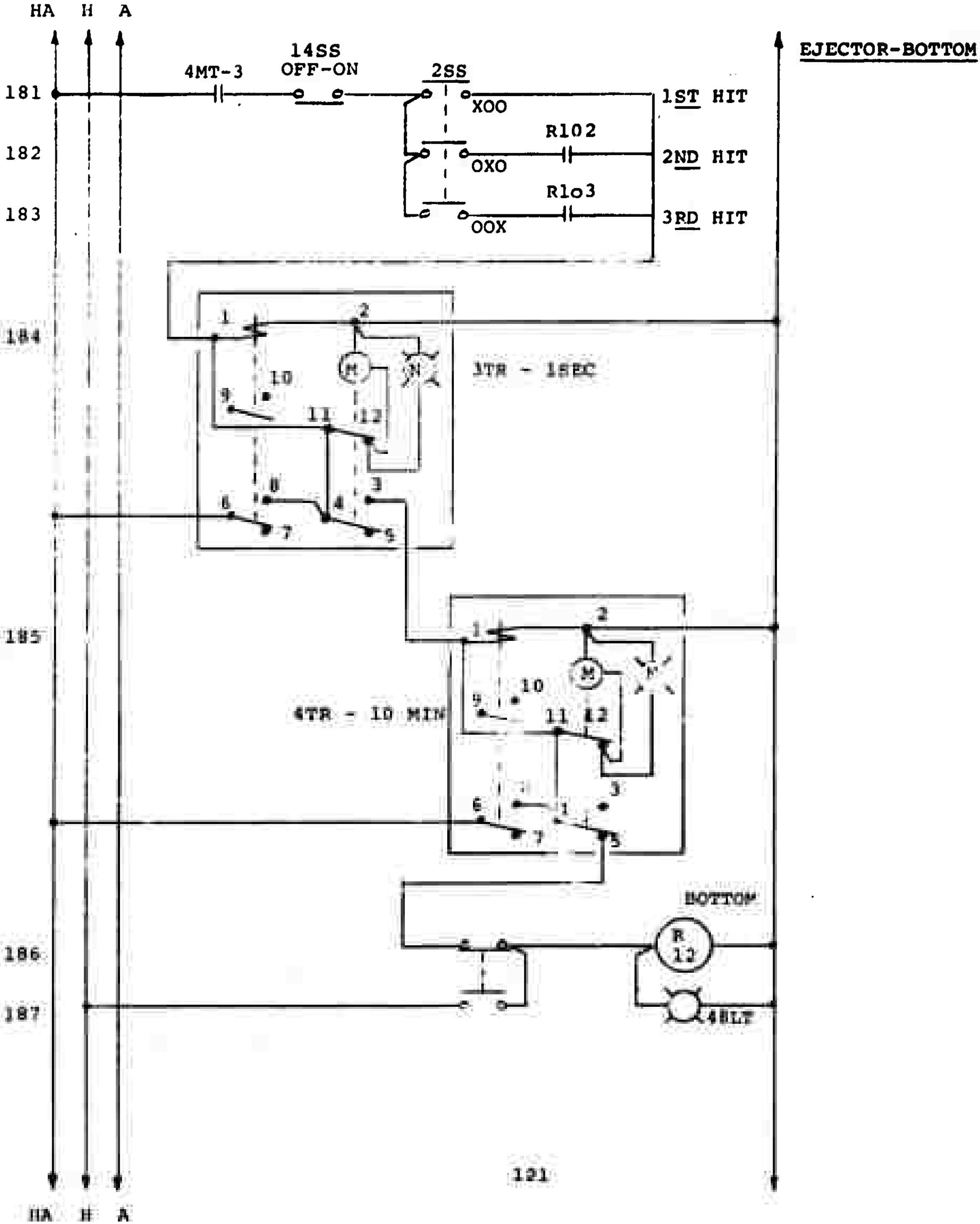
Off-on selector is turned to "on." Count selector switch is turned to 3 cycles. Thereafter:

8) On first cycle, contact 4MT-2 closes, but produces no signal, as count selector switch is open in this position.

9) On second cycle, contact 4MT-2 closes, but produces no signal, as count selector switch is open in this position.

10) On the 3rd cycle initiation, and in preparation thereto, contact R-103 is now closed. When contact 4MT-2 closes in this cycle, operation proceeds as described in paragraphs 2) through 5) above.


HA
H
A
172
173
174
175
176
177
178
179
180
4MT-2
13SS
OFF-ON
2SS
XOO
OXO
OOX
R102
R103
1ST HIT
2ND HIT
3RD HIT
EJECTOR-TOP
1TR - 60SEC
2TR - 10MIN
M
N
1
2
3
4
5
6
7
8
9
10
11
12
TOP
R
12
47LT
HA
H
A


HA H A
EJECTOR-BOTTOM
181
4MT-3
14SS
OFF-ON
2SS
XOO
1ST HIT
182
R102
OXO
2ND HIT
183
R1o3
OOX
3RD HIT
184
3TR - 1SEC
185
4TR - 10 MIN
186
BOTTOM
R
12
187
4BLT
HA H A

CIRCUIT FEATURES:

a) Accumulators two and three are each controlled by selector switches legended:

1 HIT	2 HITS	3 HITS
Off-on	Off-on	Off-on

b) Either one accumulator, or both, may be entirely eliminated from operation, with three switches "off."

c) Either accumulator 2 or 3 may operate during 1st, 2nd or 3rd cycle, or at whichever point the off-on selector is turned to "on."

d) Accumulator operation must be consistent with count selector switch, e.g., if accumulator selector for the third hit is turned to on, and count selector switch is turned to "2 hits," cycle will terminate after two cycles, and accumulator will not operate (third hit not being reached).

AUTOMATIC OPERATION
ONE HIT
EITHER ACCUMULATOR:

1) One hit off-on selector switch is turned to "on." Contact R-102 is also selective to one hit, and is normally closed during first cycle only.

2) At programmed point, contact 4MT-4 closes, energizing timer 2TD for delayed interval.

3) Timer 2TD "times out," energizing timer 3TD.

4) Relay R-19 (accumulator 2 exemplified) is energized. Relay R-19 contacts close, energizing accumulator 2 solenoid valve for period as set on timer 3TD dial.

5) Timer 3TD "times out," deenergizing relay R-19 and, in turn, accumulator 2 solenoid valve, terminating accumulator 2 operation.

6) Step switch contact 4MT-4 opens, as 4MT step switch responds to program.

Timers reset, and accumulator operation is concluded for this one cycle.

AUTOMATIC OPERATION (Cont'd)
2ND HIT CYCLE
EITHER ACCUMULATOR:

7) Two hit off-on selector switch is turned to "on." (Count selector switch is turned to 2 or 3 hits.

Contact R-103 is selective to two hits, and is normally closed in second cycle only.

8) Cycle proceeds as described in paragraphs 2) through 6), but in the second cycle only.

AUTOMATIC OPERATION
3RD HIT CYCLE
EITHER ACCUMULATOR:

9) Two hit off-on selector switch is turned to "on." (Count selector switch is turned to 3 hits.

Contact R-103 in three hit selector circuit, is selective to three hits, and is closed in third cycle only.

10) Cycle proceeds as described in paragraphs 2) through 6), but in the second cycle only.

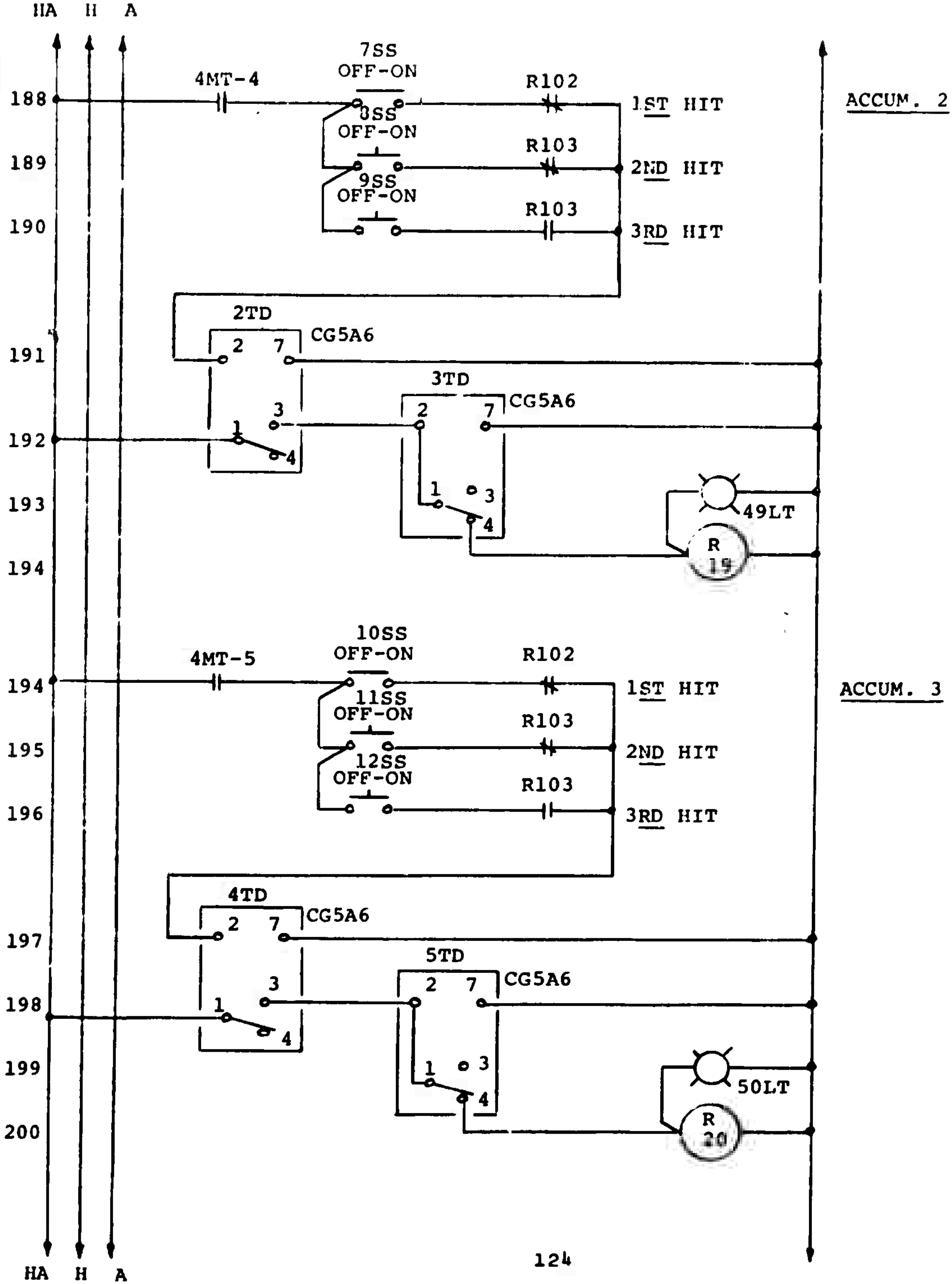
Possible accumulator operations can take place as follows:

Accumulator two:
No operation
1st cycle only
2nd cycle only
3rd cycle only.
1st and 2nd
1st and 3rd.
2nd and 3rd,
1st, 2nd and 3rd.

Similarily with
Accumulator three.

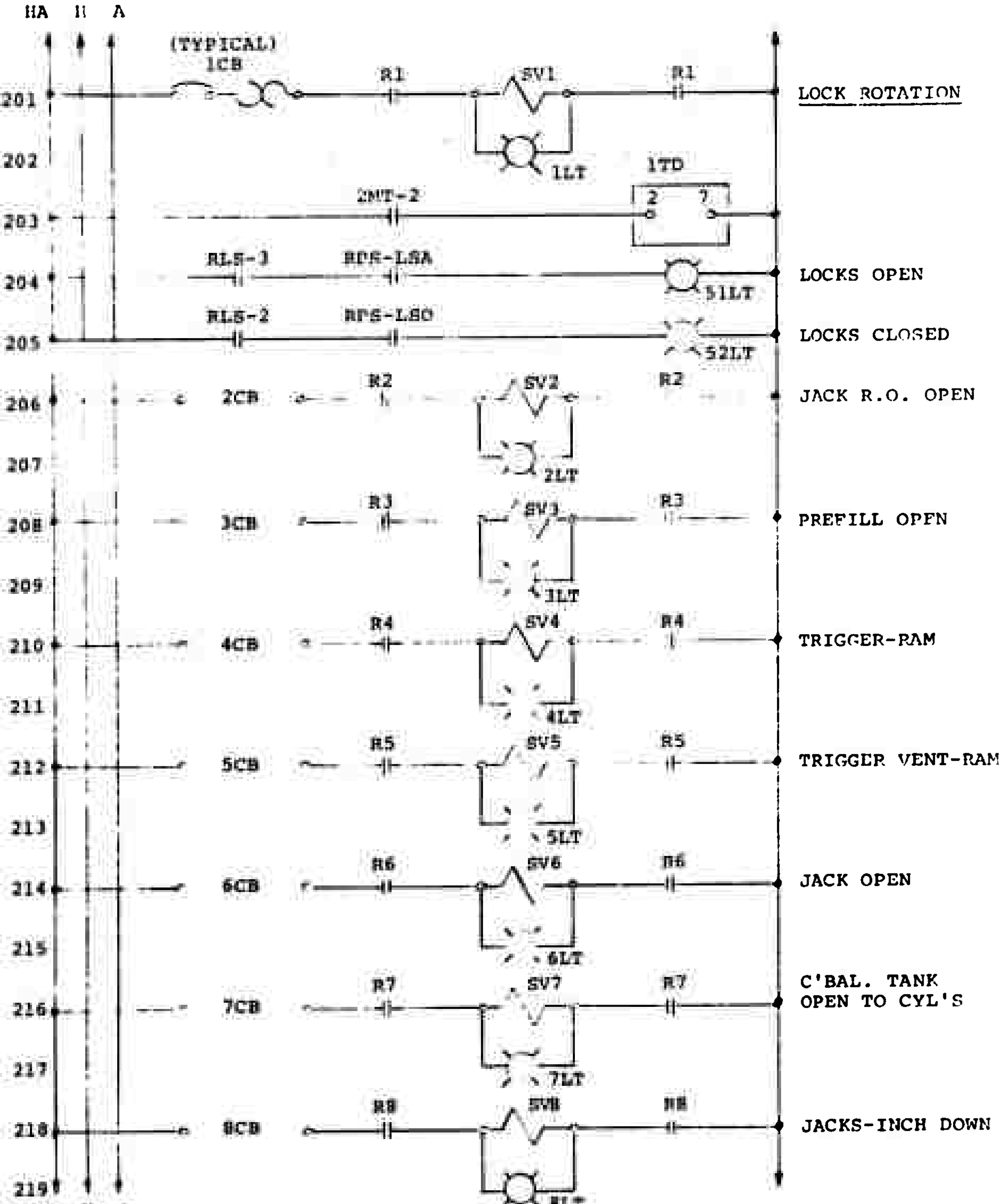
The progressive closing and opening contact sequences of relays R-102 and R103, makes the above circuit combinations possible.

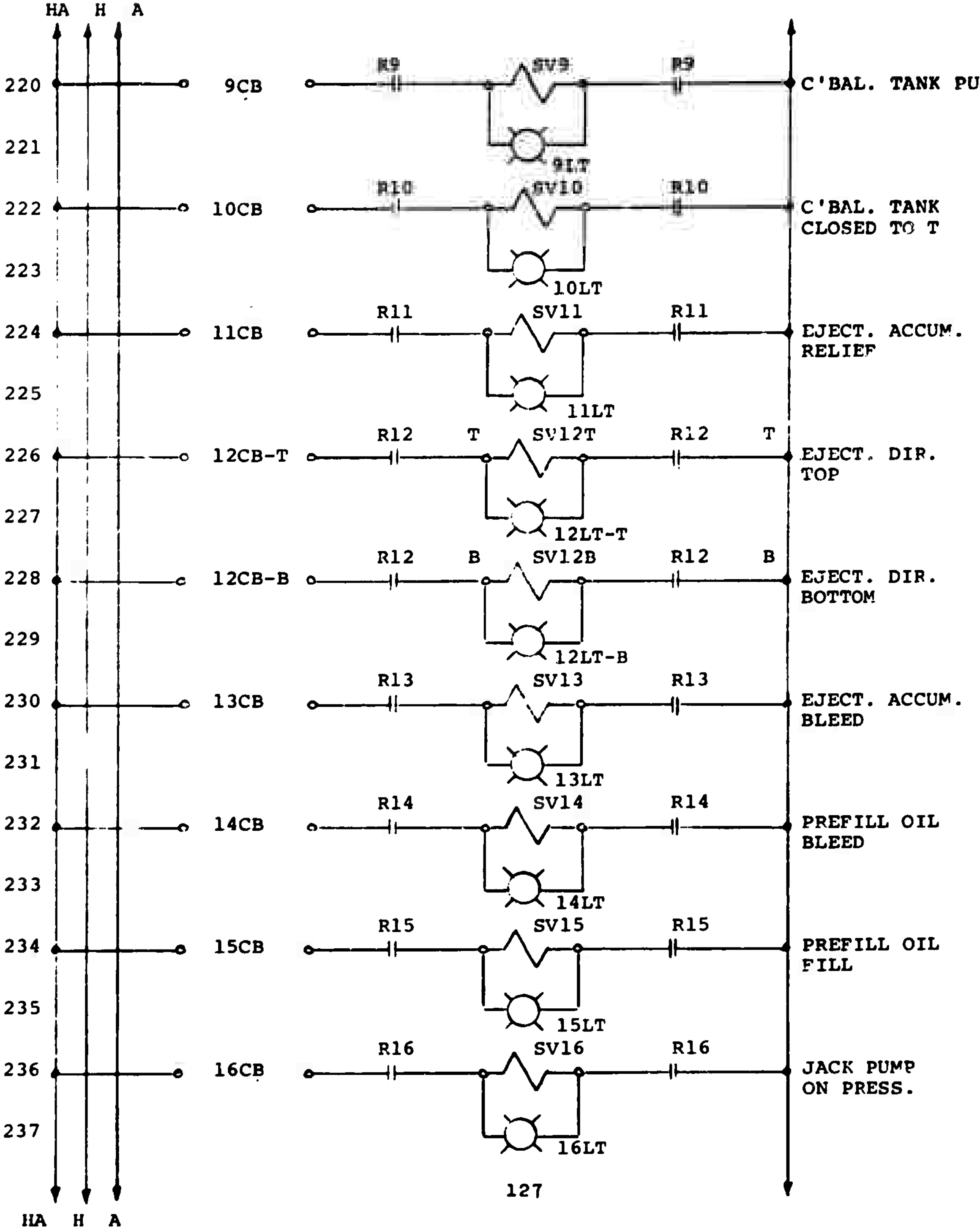
If accumulator actions is desired during 1st and 3rd cycles: Relay contact R-102 N.C. is closed during first hit only; hence no accumulator action will take place during the 2nd hit. Since relay contact R-103 N.O. will be closed in readiness for the third cycle, accumulator action will take place during the 3rd hit.

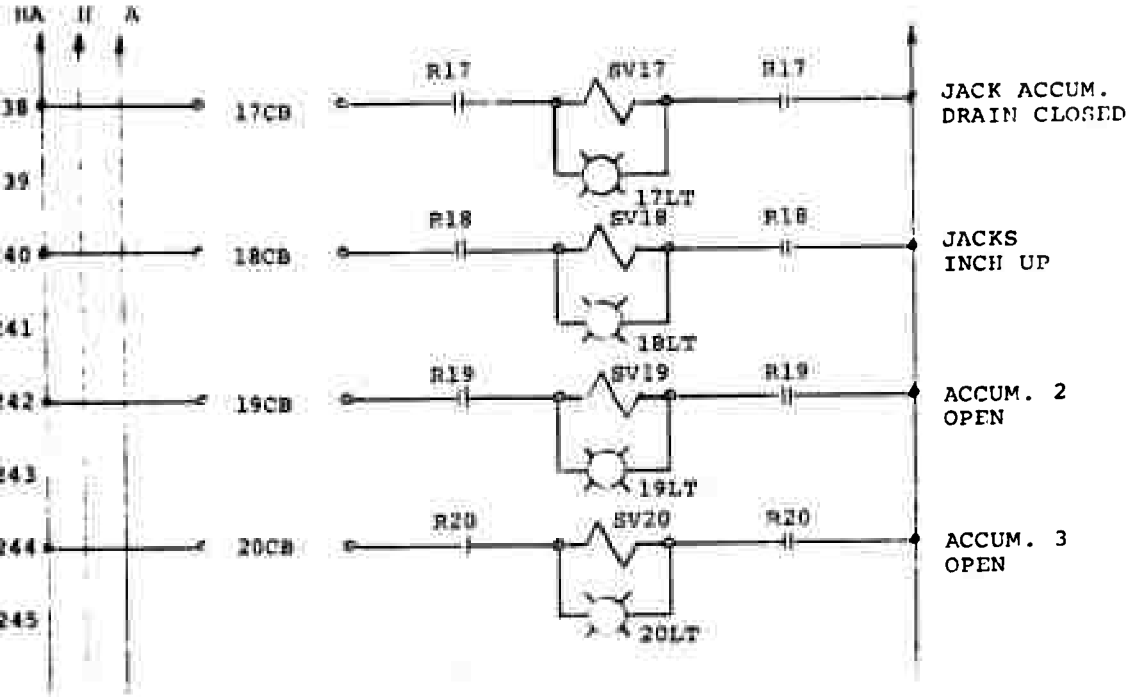

HA H A
188
189
190
191
192
193
194
194
195
196
197
198
199
200
4MT-4
7SS
OFF-ON
8SS
OFF-ON
9SS
OFF-ON
R102
R103
R103
1ST HIT
2ND HIT
3RD HIT
ACCUM. 2
2TD
CG5A6
3TD
CG5A6
49LT
R
19
4MT-5
10SS
OFF-ON
11SS
OFF-ON
12SS
OFF-ON
R102
R103
R103
1ST HIT
2ND HIT
3RD HIT
ACCUM. 3
4TD
CG5A6
5TD
CG5A6
50LT
R
20
HA H A

SOLENOID CONTROL

Note that all solenoids are protected by circuit breakers. Common numbers are used between solenoids and control relays for simplified identification.


HA H A
(TYPICAL)
1CB
R1
SV1
R1
LOCK ROTATION
1LT
1TD
2MT-2
2 7
RLS-3
RPS-LSA
LOCKS OPEN
51LT
RLS-2
RPS-LSO
LOCKS CLOSED
52LT
2CB
R2
SV2
R2
JACK R.O. OPEN
2LT
3CB
R3
R3
PREFILL OPEN
3LT
4CB
R4
SV4
R4
TRIGGER-RAM
4LT
5CB
R5
SV5
R5
TRIGGER VENT-RAM
5LT
6CB
R6
SV6
R6
JACK OPEN
6LT
7CB
R7
SV7
R7
C'BAL. TANK
OPEN TO CYL'S
7LT
8CB
R8
SV8
R8
JACKS-INCH DOWN
8LT
201
202
203
204
205
206
207
208
209
210
211
212
213
214
215
216
217
218
219
HA H A


HA H A
220
221
222
223
224
225
226
227
228
229
230
231
232
233
234
235
236
237
9CB
10CB
11CB
12CB-T
12CB-B
13CB
14CB
15CB
16CB
R9 SV9 R9
9LT
R10 SV10 R10
10LT
R11 SV11 R11
11LT
R12 T SV12T R12 T
12LT-T
R12 B SV12B R12 B
12LT-B
R13 SV13 R13
13LT
R14 SV14 R14
14LT
R15 SV15 R15
15LT
R16 SV16 R16
16LT
C'BAL. TANK PU
C'BAL. TANK
CLOSED TO T
EJECT. ACCUM.
RELIEF
EJECT. DIR.
TOP
EJECT. DIR.
BOTTOM
EJECT. ACCUM.
BLEED
PREFILL OIL
BLEED
PREFILL OIL
FILL
JACK PUMP
ON PRESS.
HA H A



HA
H
A
38
39
40
41
42
43
44
45
17CB
18CB
19CB
20CB
R17
SV17
17LT
R18
SV18
18LT
R19
SV19
19LT
R20
SV20
20LT
JACK ACCUM.
DRAIN CLOSED
JACKS
INCH UP
ACCUM. 2
OPEN
ACCUM. 3
OPEN

SELECTOR SWITCH IDENTIFICATION

1SS	MASTER MODE SELECTOR
2SS	COUNT SELECTOR SWITCH
3SS	LOCKS MANUAL
4SS	PREFILL MANUAL & JACKS RETRACTION
5SS	COMPRESSOR SELECTOR
6SS	FIRE MODE SELECTOR
7SS	ACCUM. #2 HIT SELECTION - 1ST HIT
8SS	ACCUM. #2 HIT SELECTION - 2ND HIT
9SS	ACCUM. #2 HIT SELECTION - 3RD HIT
10SS	ACCUM. #3 HIT SELECTION - 1ST HIT
11SS	ACCUM. #3 HIT SELECTION - 2ND HIT
12SS	ACCUM. #3 HIT SELECTION - 3RD HIT
13SS	EJECTOR TOP ON-OFF
14SS	EJECTOR BOTTOM ON-OFF

PUSH BUTTON IDENTIFICATION

1PB	STOP MOTION
2PB	RESUME
3PB	START
4PB	PREFILL MANUAL BLEED
5PB	PREFILL MANUAL FILL
6PB	ALARM SILENCE
7PB	MAIN COMPRESSOR MOTOR START
8PB	MAIN COMPRESSOR MOTOR START
9PB	SPLIT PUMP MOTOR STOP
10PB	SPLIT PUMP MOTOR START
11PB	FIRE

PILOT LIGHT IDENTIFICATION

1LT	1SV - LOCKS ROTATION
2LT	2SV - JACKS R.O. OPEN
3LT	3SV - PREFILL OPEN
4LT	4SV - TRIGGER - RAM
5LT	5SV - TRIGGER VENT - RAM
6LT	6SV - JACKS OPEN
7LT	7SV - C'BAL. TANK OPEN TO CYCL'S
8LT	8SV - JACKS - DOWN
9LT	9SV - C'BAL. TANK PUMP
10LT	10SV - C'BAL. TANK CLOSED TO T
11LT	11SV - EJECTOR ACCUM. RELIEF
12LT-T	12SV-T - EJECTOR DIRECTION - TOP
12LT-B	12SV-B - EJECTOR DIRECTION - BOTTOM
13LT	13SV - EJECTOR ACCUM. BLEED
14LT	14SV - PREFILL OIL BLEED
15LT	15SV - PREFILL OIL FILL
16LT	16SV - JACK PUMP ON PRESSURE
17LT	17SV - JACK ACCUM. DRAIN CLOSED
18LT	18SV - JACK INCH UP
19LT	19SV - ACCUM. #2 OPEN
20LT	20SV - ACCUM. #3 OPEN
21LT	POWER ON
22LT	ONE CYCLE
23LT	TWO CYCLE
24LT	THREE CYCLE
25LT	ONE HIT
26LT	TWO HITS
27LT	THREE HITS

28LT	READY
29LT	JACK ACCUM. LOWER
30LT	JACK ACCUM. HIGH
31LT	EJECTOR ACCUM. LOW
32LT	EJECTOR ACCUM. HIGH
33LT	COUNTER BALANCE ACCUM. LOW
34LT	COUNTER BALANCE ACCUM. HIGH
35LT	PREFILL VESSEL HIGH
36LT	PREFILL VESSEL MEDIUM
37LT	PREFILL VESSEL LOW
38LT	FAULT ALARM ACKNOWLEDGE - GRAVITY VESSEL HIGH
39LT	FAULT ALARM ACKNOWLEDGE - GRAVITY VESSEL LOW
40LT	FAULT ALARM ACKNOWLEDGE - LOCKS FAILURE
41LT	COMPRESSOR #1
42LT	COMPRESSOR #2
43LT	MAIN MOTOR STARTER
44LT	SPLIT PUMP MOTOR STARTER
45LT	PREFILL MANUAL
46LT	JACKS OPEN
47LT	EJECTOR TOP
48LT	EJECTOR BOTTOM
49LT	ACCUMULATOR #2
50LT	ACCUMULATOR #3
51LT	LOCKS OPEN
52LT	LOCKS CLOSED

RELAY CONTACT CHART

R1: 201,201

R2: 206,206

R3: 208,208

R4: 210,210

R5: 163,212,212

R6: 168,214,214

R7: 169,216,216

R8: 218,218

R9: 124,220,220

R10: 125,222,222

R11: 27,119,120,224,224

R12T: 226,226

R12B: 228,228

R13: 230,230

R14: 135, 139,232,232

R15: 234,234

R16: 25,114,115,236,236

R17: 238,238

R18: 240,240

R19: 242,242

R20: 244,244

R100: 6,22,35,53,77

R101: 8

R102: 8,9,12, 15,173,182,188,194

R103: 10,12,14,16,174,183,189,190,195,196

R104: 9,17

R105: 140,142

R106: 143,145

R107: 146,148

R108: 73,75

R109: 87,88

SEQ: 129,130,136,137

1CS: 151

2CS: 152

1MS: 154

1SPS: 156

RLS-1: 23,29,37,55,85,161,169

RLS-2: 23,37,50,78,162,170,205

RLS-3: 44,55,161,204

R-PRE-C: 134,138

R-PRE-H: 134

R-PRE-L: 134,138

RELAY CONTACT CHART

(CONTINUED)

RDP-PRE-C: 70

RDP-PRE-O: 62

RDP-R.I.C.:

RDP-R.I.O.: 81

RDP-R.O.C.: 68

RDP-R.O.O.: 60

RPS-LDC: 48

RPS-LDO: 42

RPS-LSC: 23,37,49,78,162,170,205

RPS-LSO: 43,55,161,204

RPS-PRE-C: 25,39,57,71,80

RPS-PRE-D: 64,65

RPS-PRE-O: 63

RPS-R.I.C.: 25,39,57,78

RPS-R.I.D.: 83,84

RPS-R.O.C.: 25,39,57,69,78

RPS-R.O.O.: 61

RPS-T.V.: 23,29,37,55,161,170

1MT: 8,11,13,15,34,37,53,76,164,16

2MT: 28,31,157,203

3MT: 27,30,41,67,73,74,80,158,159,

4MT: 32,41,59,87,89,166,167,172,18
188,194

SOLENOID VALVE AND FUNCTION

VALVE NO.	FUNCTION
SV 1	ROTATES TO CLEAR LOCKS
SV 2	JACK R. O. OPEN (TOP & BOTTOM)
SV 3	PREFILL OPEN
SV 4	TRIGGER RAM
SV 5	TRIGGER VENT RAM
SV 6	JACK R. I. OPEN
SV 7	C'BALANCE TANK OPEN TO CYCL'S
SV 8	JACKS INCH DOWN
SV 9	C'BALANCE TANK PUMP ON PRESSURE
SV 10	C'BALANCE TANK CLOSED TO T
SV 11	EJECTOR ACCUM RELIEF DEVENT
SV 12T	EJECTOR DIRECTION
SV 12B	EJECTOR DIRECTION
SV 13	EJECTOR ACCUM BLEED CLOSED
SV 14	PREFILL OIL BLEED
SV 15	PREFILL OIL FILL
SV 16	JACK PUMP ON PRESS
SV 17	JACK ACCUM DRAIN CLOSED
SV 18	JACKS INCH UP
SV 19	ACCUM #2 OPEN
SV 20	ACCUM #3 OPEN

APPENDIX V

Miscellaneous

I. GENERAL

A. A letter establishing liaison between General Dynamics and Lindberg Hevi-Duty, a Division of Sola Basic Industries, is attached as Enclosure No. 1.

B. Additional enclosures:

1. Enclosure No. 2 DYNAPAK Drawing 66-10 (Installation and Foundation, 1-1/2 Million Foot Pound Machine, Model No. 2436).

2. Enclosure No. 3 DYNAPAK Drawing VD-181 (Rotary Hearth Furnace).

II. ROTARY HEARTH FURNACES

A. The rotary hearth furnace is considered to present maximum flexibility of all types of furnaces considered.

B. The furnace selected is not to be considered a prototype design, since Lindberg Hevi-Duty has already designed similar furnaces, smaller in size, but of the identical configuration.

C. Future consideration should be given to evaluating the size of ingress and egress openings dependent upon the anticipated tentative forging requirements.

D. It was determined that two (2) rotary hearth furnaces would be desirable for maximum production rate.

E. Each furnace is 20'-0" in diameter and has a hearth width of 3'-2". (See Enclosure No. 3) The heating capacity of each furnace is 5000 pounds per hour.

F. Each furnace has an atmosphere control, designed by Lindberg, to reduce forging scale and to insure precision of forgings.

G. Utilizing each furnace in conjunction with the other would result in maximum output capability at the same time retaining the potential of reheating forgings if temperatures should decrease excessively. For example:

1. Assuming 200 pound billets are used, which appears to be the normal sized billet, one (1) furnace could be utilized for preheating the billets to a temperature below that which creates excessive forging scale and de-carb (approx. 950°F.

2. After preheating, the billets then would be placed in the second furnace for further heating to forging temperature (1850°F - 2150°F). Note: Less time is required to heat billets from room temperature to 950°F than from 950°F to forging temperature.

3. Therefore, with a heating capacity of 5000 pounds per hour, the following would apply:

$$\frac{5000 \text{ lb./hr.}}{200 \text{ lb. billet}} = 25 \text{ billets/hr.} - \text{prod. rate of 25 bils./hr.}$$

Note: 5000 lb./hr. is a conservative figure.

4. In retrospect, by utilizing both furnaces to heat the billets from room temperature to forging temperature, the production rate would be increased to 50 billets per hour.

LINDBERG HEVI·DUTY
DIVISION OF SOLA BASIC INDUSTRIES
[illegible] IRVING ST. • RAHWAY, NEW JERSEY 07065 • PHONE 201-388-4404

March 27, 1969

Electro Dynamic Div.
General Dynamics
Dynapak Div.
150 Avenel Ave.
Avenel, N.J.

Att: Mr. Milton Chapin

Gentlemen:

In accordance with Mr. Chapin's request, we are pleased to submit information on forge furnaces.

We are enclosing several pictures as well as drawing #62550, and these will illustrate the type of equipment appropriate for your proposed Dynapak hammer.

We would suggest that you consider the purchase of two of these furnaces. These rotary furnaces are 20'-0" in diameter and have a hearth width of approximately 3'-2". The heating capacity of each is about 5000 pounds per hour. This will total in the neighborhood of 10,000 pounds per hour heating capacity and is close to the figure we discussed.

The furnaces would be field erected and when more specific figures are available, we will work out loading and unloading methods.

The cost for the furnaces will be approximately $210,000 each.

We hope the above preliminary information will be of help to you. Please do not hesitate to contact this office for any further information we can supply you.

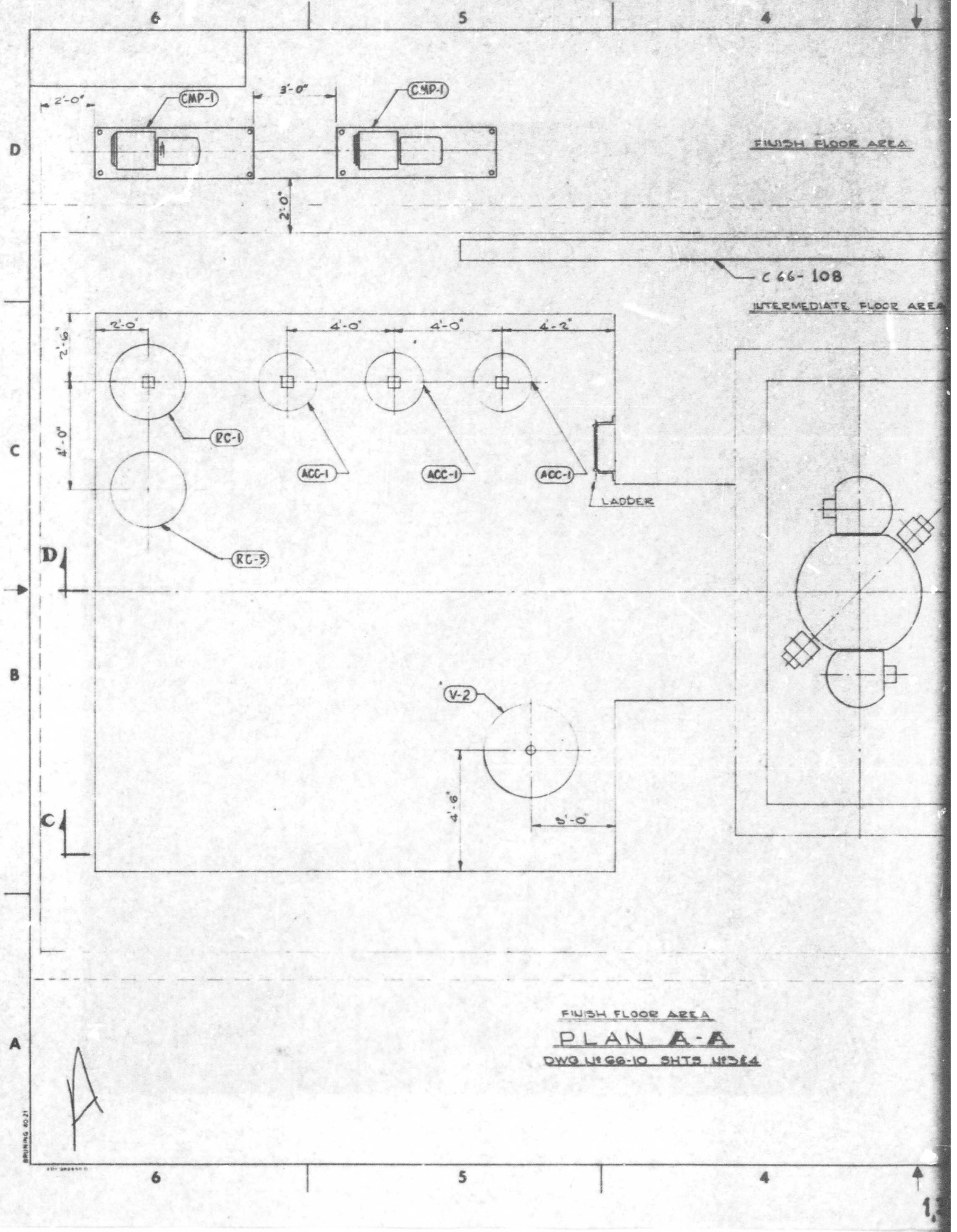
Very truly yours,

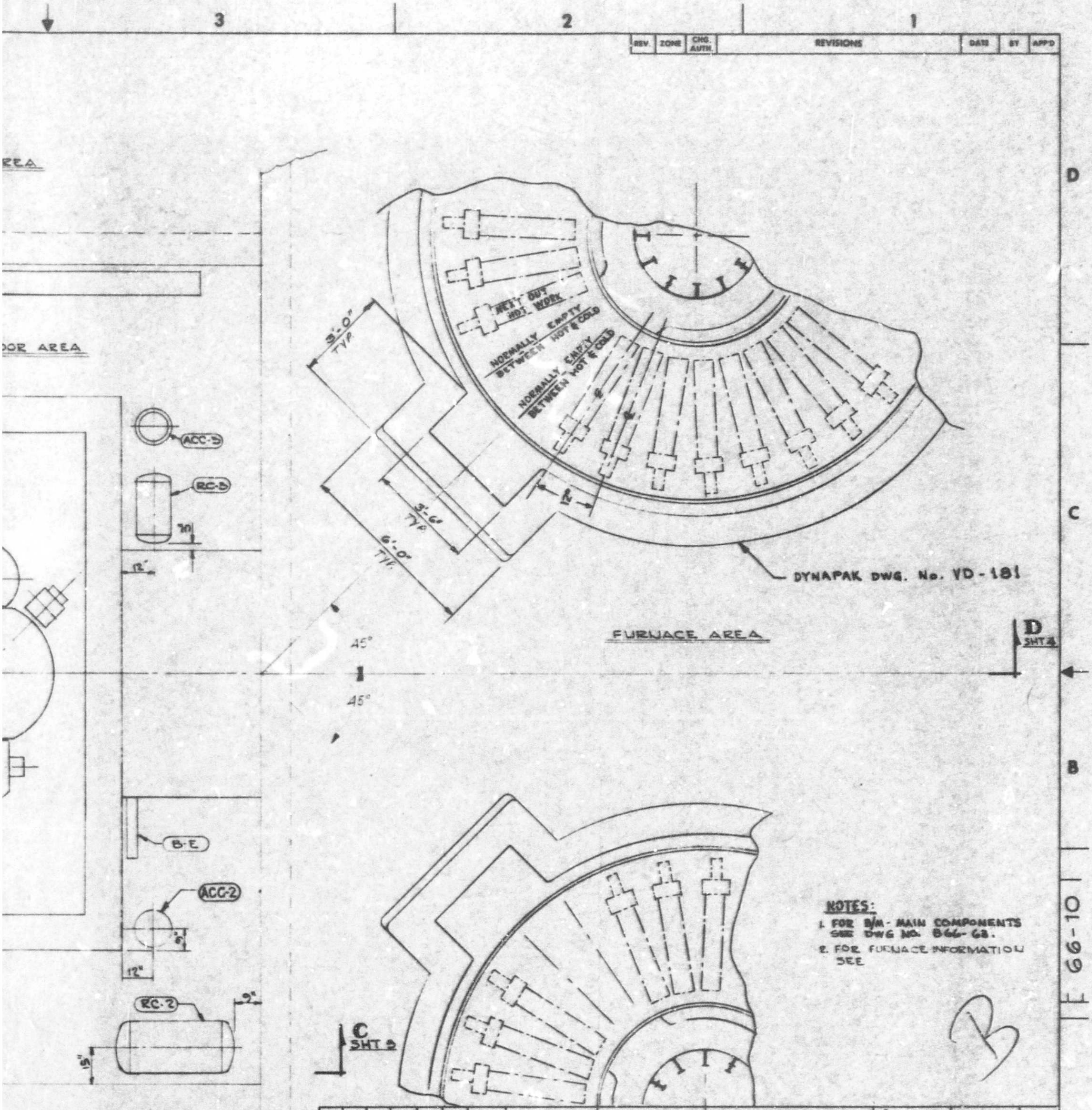
LINDBERG HEVI-DUTY
Div. of Sola Basic Industries

George Fey

138 George Fey-Sales Engineer

GF:mf (encl)


6
5
4
D
C
B
A
CMP-1
CMP-1
2'-0"
3'-0"
2'-0"
FINISH FLOOR AREA
C 66- 108
INTERMEDIATE FLOOR AREA
2'-0"
4'-0"
4'-0"
4'-2"
2'-6"
4'-0"
RC-1
ACC-1
ACC-1
ACC-1
LADDER
RC-5
D
V-2
4'-6"
5'-0"
C
FINISH FLOOR AREA
PLAN A-A
DWG № 66-10 SHTS №3&4
BRUNING 40-21
6
5
4



56	55	54	53	52	51	ITEM	PART NO	DESCRIPTION	MAT'L	MAT'L SPEC.	CODE IDENT.
						2					
						1					

REQ'D — LIST OF MATERIAL

NEXT ASSEMBLY | USED ON
APPLICATION

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES
FRACTIONS ± ANGLES ±
3 PLACE DECIMALS ±
2 PLACE DECIMALS ±

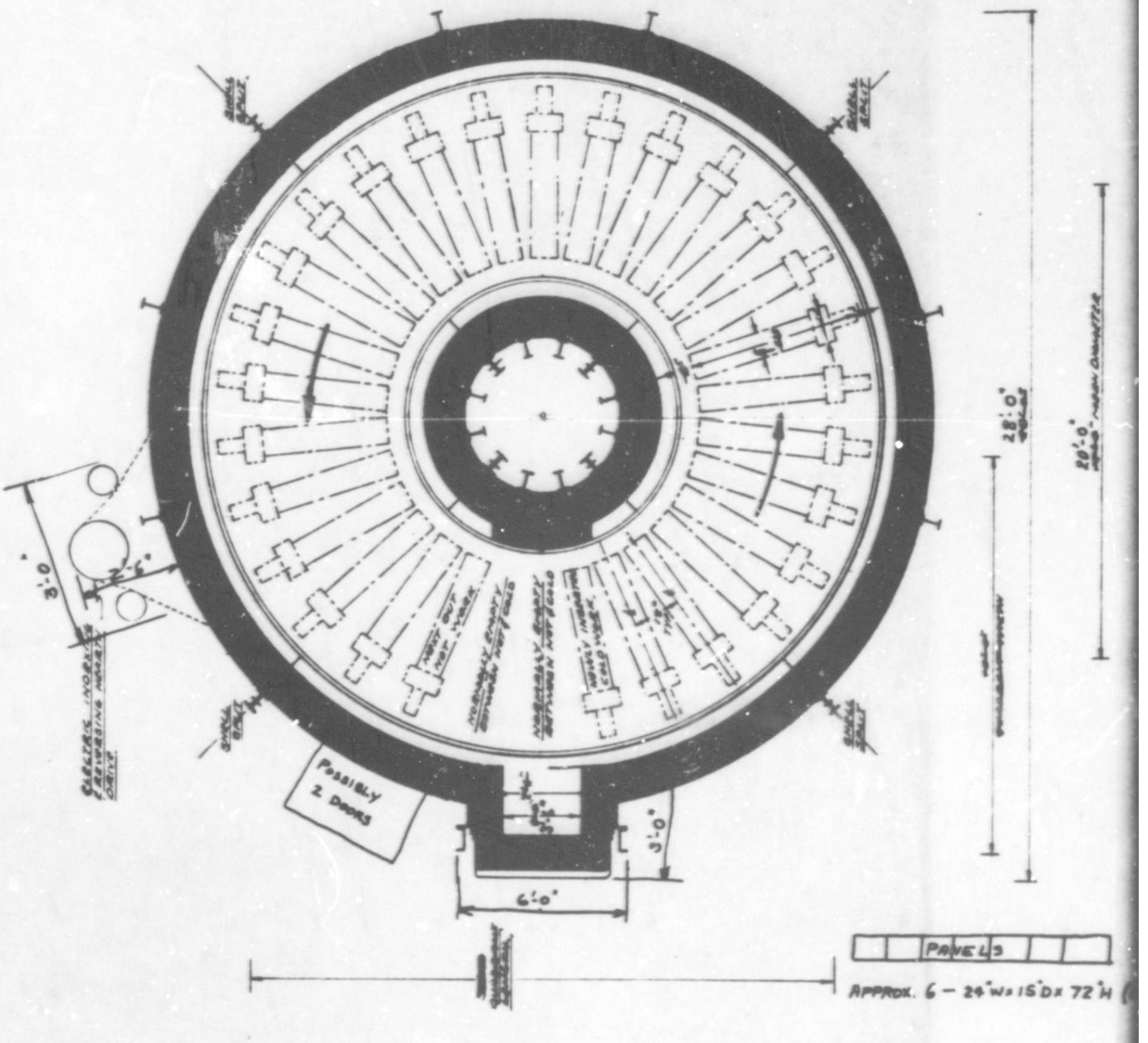
DFTM. E.T.P.
CHK.
APPD.
APPD.
REVIEWED
APPROVED
8-5-69

TITLE
DYNAPAK
INSTALLATION
& FOUNDATION
1½ MILLION FT. LB.
MACHINE MOD. 2436

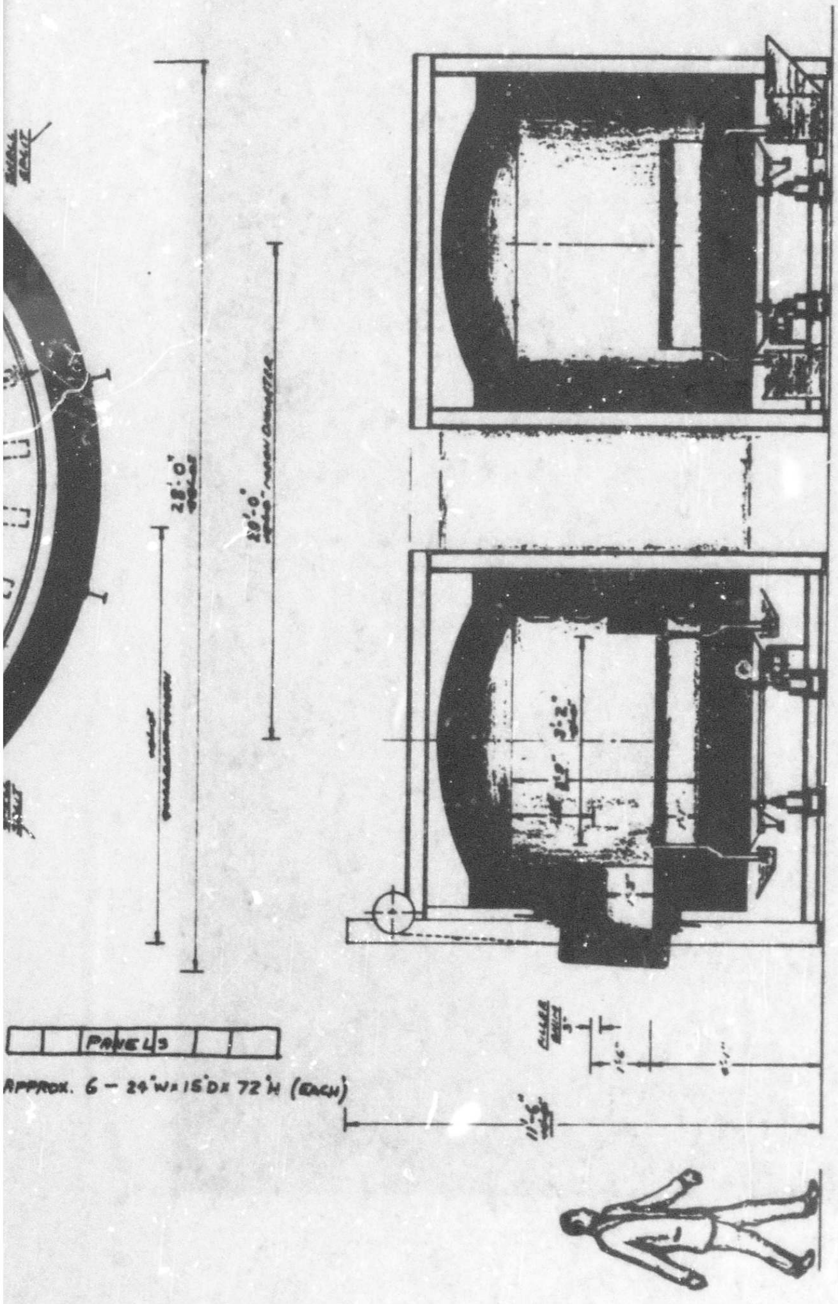
GENERAL DYNAMICS
ELECTRO DYNAMIC
AVENEL N.J.

DWG. FIRST MADE FOR | REV.

CODE IDENT 95402 | DWG SIZE D | DRAWING NO. 66-10
SCALE ½" = 1'-0" | WEIGHT | SHEET 1 OF 5



POSSIBLY
2 DOORS
6'-0"
3'-0"
28'-0"
20'-0"
PANELS
APPROX. 6 - 24"W x 15"D x 72"H



B

DYNAPAK DWG. No. VD - 181

NOTE

THIS PRINT IS ... USED FOR ... ALL DIMEN...

LINDBERG HEVI-DUTY
DIVISION OF SOLA BASIC INDUSTRIES

PRODUCT DEPT. [illegible] SPECIAL		
TYPE [illegible] ROTARY HEARTH FURNACE		
DATE JAN 29 '69	DRAWN BY [illegible]	CHECKED BY [illegible]
PRODUCT CODE 18211	SCALE 1/8" = 12"	DWG. No. 68550

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2a. REPORT SECURITY CLASSIFICATION: Unclassified

2b. GROUP: Not Applicable

3. REPORT TITLE

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4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

Final Report

5. AUTHOR(S) (Last name, first name, initial)

Chanin, Milton   Koskoris, James

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
November 1969	143	21

8a. CONTRACT OR GRANT NO.
F33615-68-C-1578

b. PROJECT NO.
TB-8-MMP-254/123-8

c.

d.

9a. ORIGINATOR'S REPORT NUMBER(S)

9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)

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11. SUPPLEMENTARY NOTES

12. SPONSORING MILITARY ACTIVITY

Air Force Materials Laboratory
Manufacturing Technology Division
Wright-Patterson Air Force Base, Ohio

13. ABSTRACT

The HERF press designed ~~under this contract~~ is a pneumatic-mechanical press of true counter-blow design. The machine incorporates mechanical, hydraulic and electrical systems. Two ~~(2)~~ integral opposing rams, weighing 73,000 lbs. each, and having a maximum impact velocity of 630 in/sec create the rated energy of 1,500,000 foot pounds. The two rams are proportioned for extreme stiffness in order to support large tooling requirements. Complete tooling interchangeability between the upper and lower rams is provided. The hydraulic portion of the pneumatic high velocity machine restores the energy released by the machine. The hydraulic system is composed of two ~~(2)~~ 100 HP motors driving four ~~(4)~~ 35 GPM pumps which provide fluid for a high performance accumulator system. This system is capable of sustaining three ~~(3)~~ successive blows of the rams within several seconds. All functions are interlocked to provide maximum smoothness of hydraulic fluid delivery with minimum inertia shock. The electrical system consists of conventional electrical circuitry monitored and controlled energy output of 1,500,000 ft.lbs. with the capability of utilizing this energy in multiple blows. The counter-blow principle and identical rams provides versatility in tooling capabilities. The composite design of the machine emphasizes simplicity, flexibility, safety, minimum maintenance requirements and increased production capability.

DD FORM 1 JAN 64 1473

UNCLASSIFIED
Security Classification

Security Classification

14 KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
High Energy Rate Forging Counter Blow Forging High Velocity Forging Controlled Energy Forging						

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